ROTARY PRESSURE EXCHANGERS

These machines are being used as superchargers in truck engines. They are equalizers—a type of dynamic pressure exchanger—whose function is to use hot pressurized gases to heat and pressurize lower energy gases. Acoustic waves scavenge and fill cells in a rotor. It seems to me that these could be of some use in an alternative air compression system.

Machinery for Direct Fluid-Fluid Energy Exc
ed. J. F. Sladky Jr., NY: ASME, 1984,

FOREWORD

In general the development of a technical capability follows several discrete steps. First, a particular phenomenon is discovered, and this, usually by accident and not as an attainment of a targeted goal. Discoveries are made almost as a by-product of some striving toward an unrelated breakthrough. The second step involves the overcoming of institutional prejudice, piecemeal and usually premature application attempts, and beginnings of true understanding of the phenomenon involved. The last step involves the injection or integration of some other technical capability heretofore unrelated to the original discovery.

Several examples can be cited from diverse disciplines. The electronic vacuum tube has now been replaced by solid state electronics which had to wait for high purity chemistry and materials in order to realize mass applications. The sophisticated aerodynamics of laminar flow airfoils are achieving their real potential only after the introduction of high strength molded composites. Thus, a technology progresses from systems that are mechanically complex and phenomenologically simple to ones that are simple mechanically and very complex in the particular phenomenon involved. Thus each technology moves from relative crudeness to extreme elegance.

Nowhere and particularly at no time as now is this more evident than in the field of fluid-dynamic machinery. The impacting technology – the computer – gives the researcher, and ultimately the designer, the power to examine a vast number of variables and to synergize very complex flowfields. Questions which, until the recent past, could only be handled by experimental and even cut and dry methods or by simplifying empiricism are now routinely addressed in the most minute detail by numerical modeling techniques. A specific case in point is the development of energy transfer devices between two fluids. In a gas turbine, for example, the combustor products are expanded in a nozzle and then enter a moving blade row of the turbine where vector velocities are changed and torque is developed. The mechanical power is then conveyed by shafting to the moving blade rows of a compressor where velocity is imparted to the fluid. The flow field is then diffused before entry into the combustor. Can this energy transfer be accomplished directly by the hot gas without the intervening complexity of turbines and compressors?

Principles of direct fluid-to-fluid energy exchange devices have been in existence for many decades. Perhaps the simplest – the shock tube – is a familiar device to students of gas dynamics. There are other devices more complex in the phenomenon involved and of considerably greater mechanical simplicity. These include the Reynst pot, Seippel comprex, Rangue-Hilsch tube, and the Crypto-Steady energy exchanger. The majority of these depend on intermittent primary to secondary fluid interaction and hence are difficult to study and control.

It is the advent of powerful computer codes that has enabled researchers to 'get into' the actual fluid-to-fluid energy exchange mechanism and its control. The development of direct fluid-fluid energy exchange machinery is poised on the brink of significant advancement. It is fortunate that some of the originators as well as major contributors of the concepts are still active in the field. It is the goal of this document to pull together under one cover information that has existed in diverse literature and to document the perspective of the original and key players in this fascinating technology. Of necessity it is impossible to represent all views of all the individuals holding them. This organizer regrets that this could not be so. However, it is hoped that this document will serve as a useful tool to students and workers in the field of direct fluid-fluid energy exchange.

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Applied Mechanics Surveys, ed. H. Norman Abramson, Harold Leibowitz 2
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PRESSURE EXCHANGE

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INTRODUCTION

Originally, "pressure exchange" meant exchange of static pressure, or, more precisely, the compression of a body of fluid by the direct action of another body of fluid expanding between the same pressure limits. Today the designation is applied to any process whereby contiguous fluid bodies or flows exchange mechanical energy through the work of mutually exerted pressure forces at their interfaces. The new definition is very broad, in that it covers also processes in which the limits of compression and expansion are widely different.

Pressure exchange cannot take place in steady flow, because no work is done by pressure forces acting on a stationary interface. Therefore, pressure exchange is always a nonsteady process.

The utilization of this mode of energy transfer is of interest because of its potentially high efficiency. In steady-flow devices for the direct transfer of mechanical energy from one flow to another, the transfer is entirely effected through the work of viscous forces and irreversible transport processes. In contrast, the work of pressure forces is essentially nondissipative, except when strong shocks are involved. Pressure exchange can, therefore, be expected to be capable of producing higher efficiencies of energy transfer than can be produced by steady-flow energy exchangers of comparable mechanical simplicity.

Early "pressure exchanger" schemes, dating as far back as 1912, appeared in gas turbine designs of Westinghouse and Bischof, where provision was made for a direct transfer of mechanical energy from the high-energy exhaust flow to the fresh charge, as a means for reducing the power consumption of the compressor (1). An arrangement proposed by Baetz in 1920 utilized pressure exchange to carry out an entire gas turbine cycle within the bucket channels of a single wheel. The combustible mixture was admitted to successive channels, where it was compressed by the direct piston action of expanding gases from other channels, and burned; and the combustion products were then brought, through stationary ducts, to compress the fresh charge in other channels (1).

Subsequent studies, for a period of about three decades, were directed along the two distinct lines of (*a*) quasi-static (or space-controlled) and (*b*) wave-controlled (or time-controlled) pressure exchange. The classic pressure exchanger of this period consists of chambers or "cells," each of which is alternately fed, at one end, the "secondary" fluid to be compressed and the "primary" fluid to be expanded. The expanding primary fluid forces the secondary fluid out of the chamber at the other end, and is then itself discharged out of it. The distribution and valving actions required to produce and control these alternate charges, interactions and discharges are obtained by mounting a number of such cells on a rotor which continually brings new cells to positions opposite appropriate stationary inlet and exit ports. Since each port finds substantially identical flow conditions at all the cell ends that come to face it, the flow through the ports is very nearly steady.

QUASI-STATIC PRESSURE EXCHANGE

Pressure exchange in the original sense of exchange of static pressure is produced in a mechanism proposed by Lebre (2). Here the expansion of a primary gas within cells on one side of the periphery of a rotor forces a "pressure exchange fluid" out of these cells, through stationary passages, into cells on the opposite side, which are occupied by a secondary gas. The stationary passages are arranged to connect the cells containing the primary gas in the earlier stages of expansion with those containing the secondary gas in the later stages of compression, and vice versa, so that pressure exchange occurs in small steps. The nonsteadiness required for the nondissipative exchange of energy is provided by the fact that the initial phase of each step of the exchange is essentially a shock tube process. However, the duration of each step is long compared to the time required for a pressure wave to travel the length of a cell. As a consequence, the average pressure in each cell is, over a relatively wide range of rotor speeds, only a function of the cell's position relative to the stationary parts of the machine. The process is said to be "quasi-static" for this reason.

In variations of the Lebre scheme, direct contact is permitted between primary and secondary fluid, and the need for a recirculating pressure exchange fluid is eliminated.

In the Poggi exchanger (3,4), the cells are arranged radially in two coaxial, counterrotating impellers. Each cell is alternately fed, through ports at the impeller eye, low-pressure secondary and high-pressure primary fluid, and alternately discharges high-pressure secondary and low-pressure primary fluid through ports around the impeller periphery. Communication between cells in the two impellers, when they find themselves back to back, is provided by openings in the impeller disks. Because of the counterrotation of the impellers, the cells which contain, in one rotor, the primary gas in the earlier phases of expansion are made to communicate in this manner with those which contain, in the other rotor, the secondary gas in the later stages of compression, and vice versa. Thus, the high-pressure primary gas is made to expand, in small steps, into confined spaces which are originally filled with the low-pressure secondary gas, in a quasi-static process which eventually brings the secondary gas to a pressure almost equal to the initial static pressure of the expanding primary gas. A complete analysis of pressure exchangers of this class has been developed by Poggi (4), with consideration of most foreseeable losses.

In an arrangement proposed by Von Ohain (5), the cells are carried by the rotor through a pressure exchange sector and through a recharge sector. In the former sector, the primary gas is admitted into the cells at one end through a succession of stationary nozzles of increasing throat area and forces the secondary gas out at the other end. In the recharge sector, the expanded primary gas is discharged from the cells and new charges of secondary gas are taken in. In the application of these exchangers as high-pressure stages in turbomachines, the cells within the pressure exchange sector form, with the combustion chamber, a "sealed room." In steady operation, the entering cells must carry, both in mass and in volume, as much low-pressure air as the leaving cells carry high-pressure hot gas. Therefore, steady operation of these devices with isobaric combustion requires that the pressure rise produced by pressure exchange be the same as that which would be produced, on the low-pressure charge, by constant-volume combustion with the same heat release.

WAVE-CONTROLLED PRESSURE EXCHANGE

In quasi-static pressure exchange the interaction takes place as a relatively slow succession of weak steps. Shock losses are, therefore, negligible, but extensive mixing and interdiffusion between the interacting fluids are unavoidable because of their long residence time in contact with one another within the exchanger, and the flow capacity is normally very low.

With fewer and stronger steps, large-amplitude pressure waves come to play a dominant role in the transfer of energy, and shock losses may become important. However, great gains in flow capacity are possible if the travel time of these waves is appropriately adjusted to the residence time of the fluid particles in the exchanger. This fact was not fully recognized until 1940, when a successful mechanism of pressure exchange based on the utilization of wave processes was proposed by Seippel, to be later embodied in the Brown Boveri "Comprex" (6–14). Here compression takes place, again within cells at the periphery of a rotor, in two stages. The first stage is provided by the hammer wave which is produced by sudden closure of the downstream end of the cell; and the second stage by the shock which is generated by the piston action of the primary fluid when it is suddenly admitted into the cell. The control of these wave processes, as well as of the subsequent purging and recharging, requires that the valve action resulting from the motion of the cells relative to the stationary ports and shields be carefully timed to the motion of waves and interfaces within the cells.

In most applications of these exchangers, the primary is made up of hot combustion products and the secondary of fresh air, and a favorable mismatching of stagnation pressures results from the fact that velocities and static pressures are the same on the two sides of interfaces separating the two gases (15). Furthermore, and for the same reason, the mass flow rate of the secondary through the exchanger exceeds that of the primary. Therefore, this type of pressure exchanger lends itself to utilization in looped arrangements as a compressor, when a portion of the secondary flow is energized through combustion, after pressure exchange, to form the primary flow; or as a gas generator, when the entire secondary flow is so energized but only a portion of the hot gas is recirculated in the exchanger. In the latter form, this device may take the place of high-pressure stages in turbomachines, the balance between work of compression and work of expansion being obtained, for this portion of the cycle, by adjustment of the pressure limits. Early workers on the Comprex achieved pressure exchange compression ratios of about 2.5 (8,9). However, preliminary performance estimates based on acoustic approximations or on the assumption of polytropic transformations in pressure exchange (12) proved to be inaccurate even for relatively low pressure ratios. Detailed analyses by the method of characteristics, first carried out by Kantrowitz and his associates (15), added much to the understanding of the operation of these machines and led to notable improvements of their performance.

Important flow distortions are produced in the rotor passages of these exchangers by the centrifugal body forces. The transverse pressure gradients associated with these forces depend on the local densities and cannot, therefore, be matched across normal shocks or interfaces. By the use of helical cells, the tangential component of the motion of the gas in the stationary frame of reference may be reduced (6) but not eliminated, because the relative velocity in the cells is not constant. In a variation proposed by Darrieus (16), the cellular assembly is stationary and gas flow and wave processes are controlled by rotating the end shields with portions of the ducts leading to the ports. Centrifugal effects are thus confined to steady-flow regions.

Neither quasi-static nor wave-controlled exchangers can normally produce the equality of pressure limits which is specified in the original definition of pressure exchange. In the former, because of the stepped character of the process, the amplitude of the pressure change is smaller in the secondary than in the primary fluid; in the latter, mismatching will normally produce the opposite effect. On the other hand, different pressure limits are desirable in some applications. In wave-controlled exchangers, the separate regulation of the pressure limits can be achieved by the generation of more complicated wave processes, through suitable mod-

ifications of the timing of valve operation (17). These modifications will normally affect the flow capacity of the device.

Pressure exchangers based on wave processes have been shown to have merit as superchargers (13,14). In their application as high-pressure stages (in looped arrangement with the combustion chamber) in turbomachines (7, 8,18), i.e., as supercharged gas generators, they have the advantage of permitting very high peak temperatures, because (except in the Darrieus arrangement) all their moving parts are alternately exposed to hot and cold gases. Furthermore, supercharging improves their flow capacity.

WAVE ENGINES AND WAVE COMBUSTORS

Unsupercharged gas generators of this class, when the discharged gases are utilized to form propulsive jets, are called "external-combustion wave engines." These are usually double-flow engines, the exhaust [illegible]omprising a low-energy and a high-energy jet. [illegible]he former is made up of the de-energized pri[illegible] as it is scavenged out of the cells, and the latter [illegible]at portion of the primary which is not recirculat[illegible]rough the pressure exchanger. Scavenging of the low-energy flow may be promoted by ejector action, with the high-energy jet as the driving flow (19). External-combustion wave engines are believed to be potentially competitive with the turbojet in air and fuel specific impulse but not in thrust per unit area (20).

A wave-controlled gas generator in which pressure exchange and combustion take place in the same cells is called a "wave combustor" or an "internal-combustion wave engine," depending on whether its exhaust is used to produce shaft power or a propulsive jet.

A simple illustration of this form of pressure exchange is provided by the shock which is generated on ignition of a combustible mixture at one end of a tube. The shock is sustained by combustion and travels ahead of the flame, acting as the vehicle for the transfer of energy from the burning gases to the combustible charge. Utilization of a similar mode of pressure exchange was attempted by Kahane in the "intermittent ramjet" (21). Higher compression ratios can be obtained by "extended pressure exchange," i.e., by suitable delays in the utilization of the explosion-generated waves. For example, some of the waves which are produced by the explosion in each cycle may be used to establish the flow conditions which will permit the generation of a strong hammer wave through interruption of the flow in the next cycle. The interruption of the flow may be produced by the rapid closure of a gate (22), or by the utilization of the pressure waves that are generated when interfaces arrive at a short converging nozzle (20,23,24). In general, these processes are so controlled and timed that the effect of each explosion is not so much to generate a new pressure wave as it is to supply new energy to selected carry-over waves from the preceding cycle. Under these conditions, the combustion-energized shocks may build up to great strength and produce a high compression ratio. On the other hand, the entropy rise which is produced in the repeated passes of these shocks may also become very large. The determination of the best compression ratio is complicated by the fact that the freedom of choice of combustion modes is far greater in nonsteady than in steady flow (25). The entropy rise associated with pressure exchange depends on the number of cycles during which each particle remains within the exchanger before it undergoes combustion and on the number and strength of the pressure exchange shocks, which in turn depend on the rate and mode of heat release and on the boundary conditions. Pangburn (26) was able to identify certain optimum combinations of shock strength and heat release, within a class of pressure exchange modes with front modes of combustion. He found that the thermal efficiency of internal-combustion wave engines is potentially high at its peak but in every case very sensitive to departures of the pressure exchange mode from the optimum.

A study by Fairhall (27) has revealed great variations in the rate at which internal-combustion wave engines approach periodic operation, depending on the engine configuration and on the initial conditions. These results cast serious doubt on the reliability of any performance estimate based on analysis of a few initial cycles by the method of characteristics.

Most experimental work in this area has been concerned with the problems of combustion and cyclic reignition by hot surfaces, and special attention has been given to the purging of the boundary layer on these surfaces to expedite their contact with the fresh charge in each cycle (24,28).

Pressure exchange is entirely of the extended form in the "hammerjet" (29,30,31,32) and in the hammer-wave combustor (33), where wave precompression of the charge is accomplished by a single sweep of a hammer wave, to be followed by combustion at substantially constant volume. The discharge of the combustion products here occurs in two phases —high-velocity expansion and low-velocity purging. The duration of the latter phase is normally the main frequency-determining factor in this type of device. Purging may be expedited by maintaining an adequate pressure drop across the exchanger (either through utilization or ram effects or by supercharging) or, in multitube devices, by the ejector action of the high-energy portion of the exhaust.

Generalized methods of analysis and performance estimates of wave engines and wave combustors, based on consideration of the entropy produced in the flow transformations within these devices are developed in (20).

PULSATING-FLOW THRUST AUGMENTATION

Wave-controlled pressure exchange is of special interest as a means of effecting the direct transfer of mechanical energy from the hot combustion products to a secondary flow of cold air in intermittent-jet engines, for the purpose of thrust augmentation (20). The spontaneous occurrence of such processes in the tailpipe of most standard pulsejets at low flight speed ("backflow augmentation"), first recognized by Schmidt (34), has been the object of several experimental studies (35,36,37). Similar processes take place in thrust-augmenting ejector arrangements in which the primary flow is nonsteady (38). Here the transfer of energy takes place by both pressure exchange and mixing, but the former effect usually predominates and theoretical analysis has shown that the energy transfer efficiency can be very high (20). The superiority of this form of augmentation over that of the steady-flow ejector has also been confirmed by experiment (39, 40). Attention has even been given to its utilization on steady-flow engines through conversion of the steady exhaust, by means of choppers, to a pulsating flow (41). Unfortunately, the primary momentum loss produced by the conversion offsets much of the thrust advantage that could otherwise be derived from the better mode of augmentation. In other arrangements that have been studied for the same purpose, pulsating augmentation is obtained without significantly disturbing the steadiness of the primary exhaust. In one of these arrangements, the augmenting duct is replaced by a cluster of ducts on the periphery of a rotating cylinder. These ducts are brought to face the stationary nozzle in succession, thereby producing in each the nonsteady flow situation that is required for wave-controlled pressure exchange. Success so far has been limited, because of the increased frictional losses. Similar effects have been achieved using a stationary cluster of ducts and a rotating primary nozzle (42).

CRYPTOSTEADY INTERACTIONS

The greatest obstacle to the analysis and successful development of wave-controlled pressure exchangers lies in the dependence of these devices on accurate timing of moving mechanical parts to wave and flow processes, and in their sensitivity to relatively minor departures from ideal conditions (noninstantaneous opening and closing of valves, diffusion of interfaces, distortion of wave fronts, etc.).

These difficulties are largely eliminated when one deals, insteads, with "cryptosteady" processes, i.e., with processes that are nonsteady but admit a frame of reference in which they are steady. Cryptosteady flows can be generated, controlled, and analyzed as steady flows in this unique frame of reference, while retaining the potential advantages of nonsteady flow in the frame of reference in which they are utilized.

Of these advantages, the most relevant to the present discussion is the ability to exchange mechanical energy nondissipatively. In the absence of body forces, Euler's equation and the definitions of total head H and specific stagnation enthalpy $h°$ yield, for a fluid element, $DH/dt = \partial p/\partial t + \bar{V} \cdot \bar{f}$ (if incompressible) or $Dh°/Dt = (1/\rho)\, \partial p/\partial t + T\, Ds/Dt + (1/\rho)\, \bar{V} \cdot \bar{f}$ (if compressible), where p, ρ, and T are the static pressure, density and temperature, respectively, s is the specific entropy, $\bar{V}$ the local velocity, $\bar{f}$ the force per unit volume due to surface viscous stresses, and t the time. These equations show that reversible transfers of mechanical energy within flow systems are possible only in regions where the local derivatives $\partial p/\partial t$ are not zero, i.e., in regions of nonsteady flow.

Consider now the pressure field associated with a nonuniform flow which is steady in a frame of reference F_1. The pressure is, of course, constant at each point fixed in space relative to F_1; but at any point that is fixed in a frame of reference F which is moving at a velocity $\bar{V}$ relative to F_1 the pressure will vary with time according to $\partial p/\partial t = \bar{V} \cdot \nabla p$. Therefore, except where ∇ p is zero or normal to $\bar{V}$, the flow in F is nonsteady—more precisely, cryptosteady—and undergoes energy exchanges that are absent in F_1. These exchanges are essentially nondissipative, because there is no entropy increment that is inherent in a change of frame of reference. In practice, the relative motion of the two frames involves some dissipation through bearing friction and the like, but this dissipation can be made negligible through proper design.

The mechanism whereby energy is added to a flow in a compressor or is extracted from it in a turbine is precisely of this type (43). For example, the flow through the rotor of an axial-flow turbine is steady and isoenergetic in the rotor-fixed frame of reference; but in a casing-fixed frame of reference the same flow is cryptosteady and yields energy to the environment. The essentially nondissipative character of this energy extraction is demonstrated by the fact that the adiabatic efficiency of the turbine could, ideally, be 100%. In contrast, the ideal adiabatic efficiency of its steady-flow counterpart—the Tesla friction wheel—is only 50%.

A pressure exchange situation arises when the system considered involves two or more flows (44). Two contiguous streams which do not exchange energy in a frame of reference F_1 in which they are both steady will, in general, exchange energy in every other frame. As in all other forms of pressure exchange, the mechanical energy exchanged between contiguous flows in a cryptosteady interaction is equal to the work done by the pressure forces which the interacting flows exert on one another at their interfaces. This work is zero in F_1, where the interfaces are stationary. In every other frame of reference the interfaces move and energy is exchanged. An important distinguishing feature of cryptosteady pressure exchange is the existence of a frame of reference F_1 in which the interfaces are stationary and the flows are therefore tangential to them on both sides: as they come in contact, the interacting flows deflect each other to common orientations at their interfaces in F_1.

In a simple embodiment of this concept, frame F_1 rotates at a constant angular velocity relative to frame F. A driving (primary) fluid is made to issue through slanted orifices on the periphery of a rotating member which is driven by the reaction of the issuing jets themselves. At every instant, the primary fluid which has emerged during a brief and immediately preceding time interval from each rotating orifice occupies a spiral or helical region in space, which rotates about the same axis and at the same angular velocity as the rotor. Although the fluid particles within this region do not follow the same motion, its boundaries are the interfaces separating the primary from the surrounding (secondary) fluid, and their relation to the flow of this secondary fluid is therefore substantially the same as that of blade or vane surfaces of the same shape.

rotating at the same angular velocity. Therefore, the interacting flows exchange energy in a stationary frame by a mechanism which is essentially similar to that of turbopumps or turbopropellers, although the "blades" are now patterns rather than bodies of abiding material. The energy acquired by the secondary fluid in this process is extracted entirely or partially from the primary flow, depending on whether the rotation of the rotor is produced and maintained solely by the reaction of the issuing jets or also by other means.

Here again, as in the pulsating-flow ejector, a portion of the mechanical energy transfer takes place by pressure exchange and the remainder by mixing. The portion transferred in the pressure exchange phase—the phase in which the interacting flows deflect each other to a common orientation in F_s—approaches 100% of the total as the relative velocity of F_s and F is increased indefinitely. When this velocity is zero, the pressure exchange phase is absent and the device is reduced to a steady-flow ejector. In all other cases, its efficiency of energy transfer is higher than that of the ejector, by virtue of its nondissipative pressure exchange phase.

The velocities of the deflected flows have different orientations on the two sides of their interfaces in every frame of reference except F_s. This fact may be utilized to separate the two flows after their nondissipative interaction, before mixing between them has made much progress (44). Because of the steadiness of the flow pattern in F_s, this separation can also be effected by means of F_s-fixed outflow ports.

Plane-flow analyses of cryptosteady pressure exchange, with or without subsequent mixing, have been worked out for special cases (44–49). Hohenemser (48) has also developed an analysis of the deflection phase based on application of an approach similar to the strip method of propeller theory. Experimental work has been reported by Guman (50), Vennos (51), Hohenemser (49), and Waters (52). Unpublished work by Farrell and Bursik at Rensselaer Polytechnic Institute and, more recently, by Grillo at Politecnico di Torino has been aimed at the generation of rotary "fluid blade" patterns for cryptosteady pressure exchange, through modified and controlled forms of rotating stall through stationary cascades.

Attention has also been given to a cryptosteady interaction of a reverse form, whereby an initially homogeneous stream is divided into two separate flows at different energy levels (53). The flow field is stationary in a frame of reference F_s, and the separation is effected by extracting the two portions of the flow through F_s-fixed discharge ports having different orientations. If F_s is kept in steady motion relative to another frame of reference F, then the flow is cryptosteady in F and energy is transferred in it from one portion to the other of the original stream through the work of the mutually exerted pressure forces at their surface of contact. The process lends itself to applications in heating, cooling, air conditioning, and pumping apparatus. Its only steady-flow counterpart is the highly dissipative mechanism of the Ranque-Hilsch tube.

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Machinery for Direct Fluid-Fluid Energy Exchange
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CRYPTOSTEADY MODES OF DIRECT FLUID-FLUID ENERGY EXCHANGE

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ABSTRACT

Cryptosteady-flow energy exchange is defined, illustrated by simple examples, and compared to other forms of direct fluid-fluid energy exchange. Areas of application are briefly described, with special attention to thrust augmentation and "external" energy separation.

NOTATION

A cross-sectional area

A_p total discharge area of primary nozzles

$A_{p'}$ nozzle exit area that would be required if primary were discharged at ambient static pressure

c flow velocity in F_s

F_s steady-flow frame of reference

F_u laboratory frame of reference

h^* specific stagnation enthalpy in F_s

h^o specific stagnation enthalpy in F_u

$\dot{m}$ mass flow rate

u fluid particle velocity in F_u

$u_{p'}$ primary velocity in F_u on isentropic expansion to ambient static pressure.

U $u/u_{p'}$

V velocity of F_s relative to F_u

$\alpha = A_p/A_e$

$\alpha' = A_{p'}/A_e$

δ $\Delta/\frac{1}{2}\dot{m}_p u_{p'}^2$

Δ dissipation (overall loss of availability in the energy exchange)

$\mu = \dot{m}_s/\dot{m}_p$ (in flow induction) or $\dot{m}_h/\dot{m}_c$ (in energy separation)

Φ = static thrust augmentation ratio (referred to the thrust of the isentropically expanded primary alone)

Subscripts

c "cold" output

e augmenter exit station

h "hot" output

p primary fluid or flow

p' primary conditions on isentropic expansion to ambient pressure

s secondary fluid or flow

THE CONCEPT

Most modes of direct fluid-fluid exchange involve both a dissipative and a nondissipative component [1][1]. The latter component is provided by "pressure exchange" -- the work of interface pressure forces[2].

For these pressure forces to do work, the interfaces must move. Therefore, pressure exchange can take place only where the interacting flows are nonsteady[3]. The introduction of nonsteadiness in direct fluid-fluid energy exchange processes is indeed of interest primarily because of its relevance to the possibility of generating and utilizing this nondissipative component of energy transfer. It does not necessarily follow, however, that the nonsteady modes are always to be preferred to the steady ones. In the first place, interface pressure forces may happen to act, in

[1] Numbers in brackets designate references at end of paper.

[2] "Pressure Exchange" originally meant exchange of static pressure -- the compression of a body of gas by the direct action of interface pressure forces exerted on it by another body of gas expanding between the same pressure limits [2,3]. The same designation has also been applied, after Seippel [4] to similarly direct exchanges of stagnation pressure between flows, and is now widely taken to mean any fluid-fluid energy exchange resulting from the work of interface pressure forces.

[3] The fact that a nondissipative transfer of mechanical energy can take place only in nonsteady flow has also been pointed out, in a different context, by R.C. Dean [5].

some modes, in a direction to oppose, rather than to promote, the desired transfer of energy; and in the second place, the nonsteadiness needed for pressure exchange does not usually come free. In many cases, the entropy cost of generating the required nonsteadiness may be large enough to outweigh any advantage that might otherwise be derived from the subsequent utilization of pressure exchange. This is well illustrated by the energetics of the pulsating-flow ejector [6], a flow-induction device in which the induced flow is driven by the surface forces that are exerted on it by pulse-generated "pistons" of the driving fluid (Fig. 1b). Ideally, flow induction takes place in this mechanism by pressure exchange alone, and mixing plays no role in it. Even in this ideal situation, however, its overall efficiency of mechanical energy transfer is relatively low, because a considerable portion of the available energy becomes trapped, for eventual dissipation, as kinetic energy of rotation within the "pistons" themselves as they are being formed [1,7]. In other modes, the cost of generating nonsteadiness may also be traced to a variety of other causes, such as the throttling and shock losses that are often associated with valve action, or dissipation in starting vorticity and wakes.

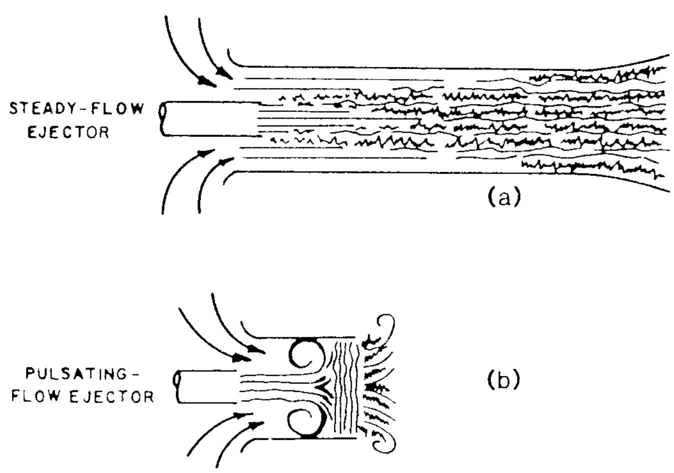

Fig. 1 - Schematics of steady-flow and nonsteady flow ejectors [6].

Special attention will be given here to one particular class of nonsteady modes -- that of the processes that are called "cryptosteady" to signify that they are nonsteady but admit a frame of observation in which they are steady [8]. These modes utilize the fact that a flow that is not uniform throughout can be steady in no more than one frame of reference. They transform a steady-flow interaction into a nonsteady one by the simple artifice of utilizing it in a frame of reference F_u other than the unique one, F_s, in which it is steady. Thus, they are capable of providing in F_u the benefit of pressure exchange, while retaining in F_s the control advantages of steady flow. Furthermore, since changes of the frame of reference are reversible, no dissipation is inherently involved in the generation of this mode of pressure exchange. To put it another way, entropy being a state function, the entropy rise is an invariant with respect to changes of the frame of reference, and is therefore the same in F_u, where pressure exchange is utilized, as in F_s, where, the flow being steady, pressure exchange is absent.

EXAMPLES

Three simple examples of cryptosteady pressure exchange, all concerning plane-flow situations, will be discussed here for the purpose of illustration.

In Fig. 2, two flows are seen deflecting each other to a common orientation in a frame of reference F_s in which they are both steady. The static pressures at a, b, and c are assumed to be equal. Neglecting transport processes, no energy is exchanged between the two flows in F_s during this "mutual deflection" phase. On the other hand, the changing magnitude of the velocity vectors in the frame of reference F_u

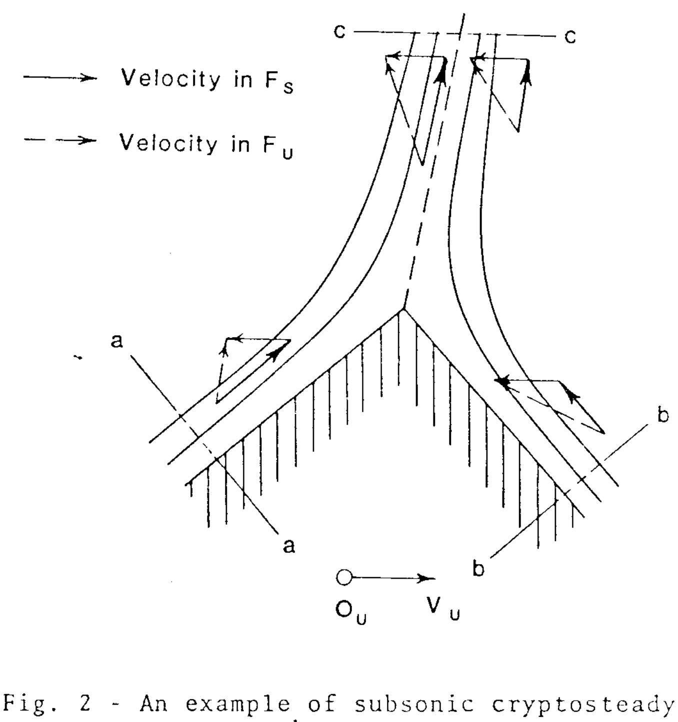

Fig. 2 - An example of subsonic cryptosteady energy exchange.

of an observer O_u moving at a velocity $\vec{V}_u$ relative to F_s reveals that a transfer of energy takes place from one flow to the other in F_u. The energy so transferred is equal to the work done in this frame of reference by the pressure forces that the two flows exert on one another at their interface. This work is, of course zero in F_s, where the interface is stationary. In short, what is a

mutual deflection of two isoenergetic flows in F_s is an energy transfer process, by pressure exchange, in F_u. It will also be noted that, whereas in this phase the velocities of the two flows acquire the same orientation in F_s, the fluid particle velocities in F_u do not.

Fig. 3 (from Ref. 11) illustrates the transformation of a steady-flow interaction into a cryptosteady one in a situation involving losses. A half-infinite, infinitely thin plate separates a high-pressure "primary" gas 1 from a low-pressure "secondary" gas 2.

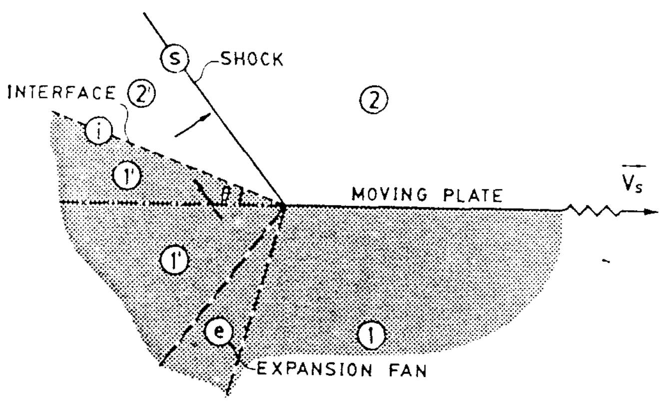

Fig. 3 - An example of supersonic cryptosteady energy exchange.

Both gases are initially (regions 1 and 2) at rest in an inertial frame of reference F_u, and in this frame of reference the plate moves in its own plane at a constant velocity $\vec{V}_s$, supersonic relative to both gases. Viscous stresses on the plate are assumed to be negligible. Aft of the trailing edge of the plate, gas 1 expands into the domain initially occupied by gas 2. At every instant, the primary gas that has expanded in a short immediately preceding time interval is seen to occupy a wedge-shaped region, the leading edge of which advances in F_u (hence relative to the gas in region 2) at the velocity $\vec{V}_s$. The frame of reference attached to the plate is in this case the steady-flow frame F_s. The two flows undergo in F_s the same turning angle θ, the primary through a Prandtl-Meyer expansion e, and the secondary through an oblique shock s. To the extent that transport processes can be neglected in the early phases of the interaction, both flows are, in F_s, isoenergetic. In F_u, however, they are not, because both the static enthalpy and the kinetic energy in F_u are higher in region 2' (behind the shock, hence after deflection) than in region 2. Since no work is required to move the plate, it follows that, in F_u, the energy acquired by the secondary gas is energy extracted from the primary gas. The energy level of the primary gas in F_u is indeed found to be correspondingly lower in the expanded region 1' than in region 1. The only entropy increment in the process is that produced by the shock. It is an invariant, hence the dissipation is the same in F_u as in F_s. Again it will be noted that pressure exchange causes the particle velocities of the two fluids to acquire different orientations in F_u. This observation has suggested certain methods of interaction control that are of potential interest in cryptosteady-flow thrust and power generation (see THE ROTARY JET, below).

A third example of cryptosteady pressure exchange is seen in Fig. 4, which shows a plane and initially uniform jet j issuing from a nozzle and impinging at an angle on a fixed wall. F_s is in this case the frame of reference attached to the nozzle. The impingement causes the jet to divide into two subflows, c and h. The stagnation stream surface

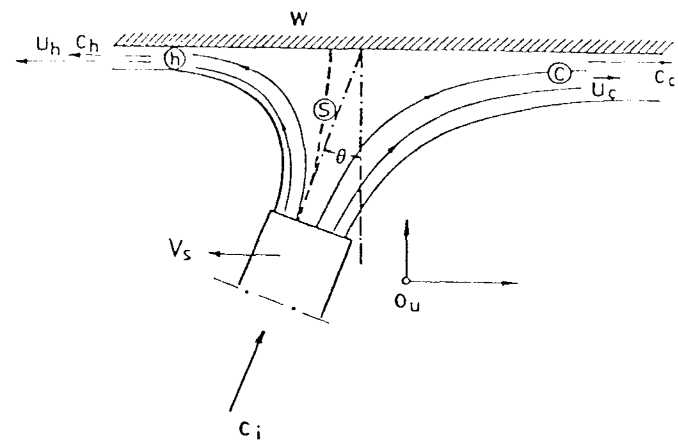

Fig. 4 - Another example of cryptosteady-flow energy exchange

s is the surface of contact between the two subflows in the impinging stream. If viscous stresses and heat exchanges are negligible, and if the static pressure is the same in the deflected subflows as in the original stream, the specific stagnation enthalpy (or the total head, if the fluid is incompressible) is, in frame F_s, the same in the deflected subflows as in the original stream. In every other frame of reference, however, the two subflows will acquire different energy levels as they become separated from one another. For example, if the nozzle is moving to the left relative to a frame F_u, an observer in the latter frame will measure a higher velocity, hence a higher kinetic energy and a higher overall energy level, in h than in c. The fact that energy is transferred, in F_u, from subflow c to subflow h can again be explained as resulting from the fact that interface s is moving to the left in F_u and that, as a consequence, the pressure forces that flows c and h exert on one another at their contact surface are doing positive and negative work, respectively. The energy that is transferred in F_u is indeed equal to the work done by subflow c on subflow h in this frame of reference.

APPLICATIONS AND THEIR RATIONALE

Cryptosteady modes of direct fluid-fluid energy exchange find application in two main

areas -- flow induction and energy separation. In flow induction the object of the exchange is the direct transfer of available mechanical energy and/or momentum of coherent motion from a "primary" to a "secondary" flow -- e.g., for the purpose of thrust of lift augmentation. In energy separation, on the other hand, an initially homogeneous flow is split into two subflows, in one of which the energy level is increased at the expense of the energy level in the other (as in the situation of Fig. 4), and the two outputs may be used either jointly or separately in environmental control or for other purposes[4].

In both areas of application, the motion of F_s relative to F_u is most conveniently effected as a rotation, frame F_s being attached to a rotor and F_u being the laboratory frame of reference. The rotor receives the appropriate flow (in flow-induction, the primary flow; in energy separation, the flow to be split) and discharges it through canted nozzles on its periphery, thereby being driven by the reaction of the issuing jets themselves. Energy separation is called "external" or "internal", depending on whether it takes place outside or inside the rotor. Figs. 5(a) and 5(b) show a radial and an axial flow induction arrangement, respectively, and Figs. 6(a) and 6(b) show two energy separation arrangements -- an external and an internal one, respectively.

As already noted, the special merit of cryptosteady modes derives, in both areas of application, from two factors; (a) the role that is played in them by the nondissipative mechanism of pressure exchange, and, no less importantly, (b) the uniquely nondissipative way in which they generate pressure exchange.

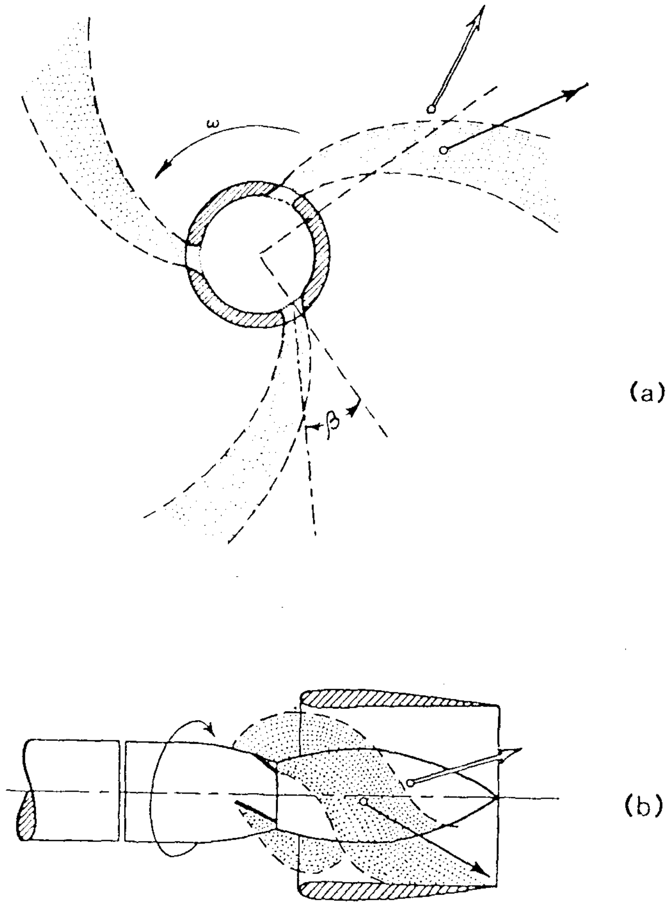

Fig. 5 - Cryptosteady flow induction
(a) Radial (b) Axial

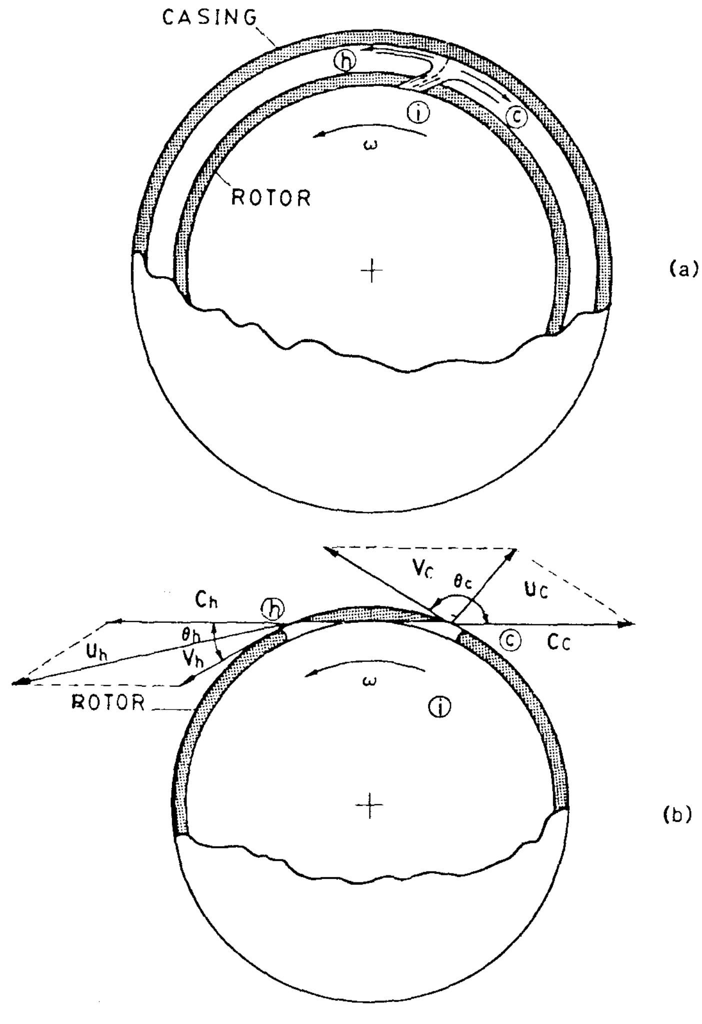

Fig. 6 - Cryptosteady flow energy separation
(a) external, (b) internal

Strangely enough, the importance of the effect of dissipation on performance has not always been fully appreciated. As an illustration of this effect, consider a thrust-augmenting ejector of given area ratio A_e/A_p' operating statically on incompressible fluids of equal densities. The flow induction process may be steady or nonsteady or

[4] Flow induction devices and energy separators are sometimes called "equalizers" and "dividers", respectively [9,10].

cryptosteady. If, however, the flow at the exit is assumed to be steady and uniform (fully mixed) in all cases, and if it is further stipulated that no net energy be exchanged between the internal flows and the surroundings, it can readily be shown, from the definitions of δ and Φ_0, that

$$\delta = 1-(1+\mu)U_e^2 \quad (1)$$

$$\Phi_o = (1+\mu)U_e \quad (2)$$

and, by continuity,

$$U_e = \alpha'(1+\mu) \quad (3)$$

Therefore,

$$U_e = [(1-\delta)\alpha']^{1/3} \quad (4)$$

$$\mu = [(1-\delta)/\alpha'^2]^{1/3}-1 \quad (5)$$

and $\Phi_o = (1+\mu)^2\alpha' = [(1-\delta)^2/\alpha']^{1/3}$ (6)

Eqs. (5) and (6) are plotted in Figs. 7 and 8, respectively. They show that, with any given area ratio, μ increases as δ is decreased, and that, as a direct consequence of this effect, impressive performance advantages can be derived from the use of the least dissipative modes of flow induction. They also show that a comparison of the performance merits of different ejector mechanisms for equal mass flow ratios could be misleading, in that, if the area ratio is also specified (as is often the case), δ is of course found to be uniquely determined, regardless of the mode of flow induction. Results of such comparative analyses have indeed been misinterpreted by some (obviously unaware of clear experimental evidence to the contrary) to mean that the only beneficial effect that can be derived from the introduction of nonsteadiness in the ejector mechanism is the promotion of mixing [12 through 16][5].

Similar and even more obvious arguments may be presented in support of the application of cryptosteady modes in energy separation. All else being equal, a lower dissipation in the energy separation process means a higher availability in the outputs (mostly notably in the "hot" output), hence the potential for high second-law efficiencies.

In conclusion, the best mechanism of direct fluid-fluid energy exchange are those which utilize pressure exchange, and in which pressure exchange itself is achieved at the least entropy cost. The cryptosteady modes are believed to be among the most promising of such mechanisms.

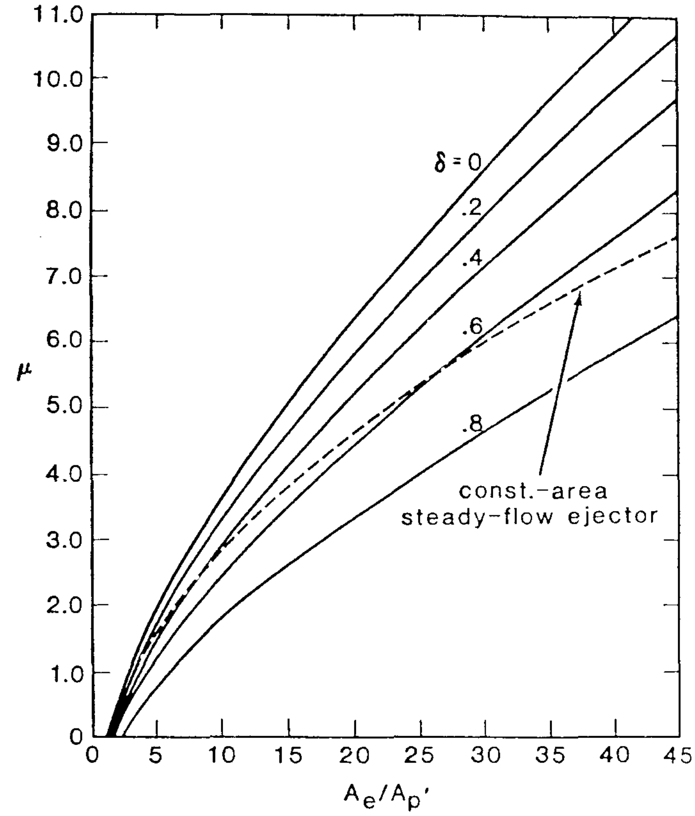

Fig. 7 - Effect of dissipation on entrainment ratio

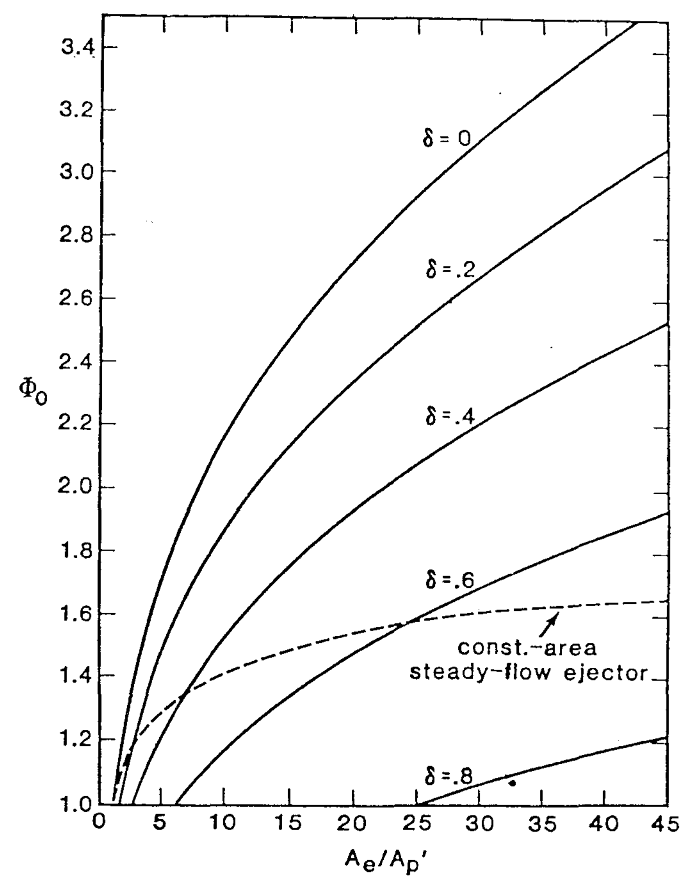

Fig. 8 - Effect of dissipation on thrust augmentation ratio

5 The misinterpretation stems, of course, from failure to recognize that the problem of flow induction is a boundary-value problem. It is noteworthy, in this connection, that significant error may also result from initial-value stipulations concerning only one of the two flows [1].

THE ROTARY JET

The flow-induction arrangement of Fig. 5(b) is known as the "rotary jet" [11, and 17 through 28]. In this device, as already mentioned, the primary is discharged into the interaction space through skewed nozzles on the periphery of a free-spinning rotor, thereby causing the rotor to spin. The inclination of the nozzle axis to the local meridional plane is called the "spin angle". Of course, the higher the spin angle -- everything else being the same -- the higher the rotor speed. The interaction is steady in a frame of reference fixed to the rotor (F_s) and nonsteady in every other frame of reference. At every instant, the primary fluid that has emerged in an immediately preceding time interval from each rotating orifice occupies in space a spiral or helical region, called "pseudoblade," which rotates about the same axis and at the same angular velocity as the rotor. Although the fluid particles within the pseudoblade do not follow this same motion, its boundaries are the interfaces separating the primary from the secondary fluid, and their motion gives rise to pressure exchange. The secondary acts on the primary in the manner of a turbine, while, at the same time, the primary acts on the secondary in the manner of a propeller or fan. The two actions are compounded here in a single step, whereby, as in the situation of Fig. 2, the two flows deflect each other to a common orientation in F_s and, as a consequence, pressure exchange takes place between them in F_u. The ways these actions are finally utilized in rotary jets differ, of course, from the ways of dynamic flow machines, because the pseudoblades are fluid[6]. For example, in rotary-jet propulsion, the thrust cannot be generated as a force exerted by the pseudoblades on the body to be propelled. It is generated instead, as the resultant of the surface forces that the inducing and the induced flow are exerting on the boundaries of the interaction space (and most particularly, as in ducted propellers, on the shroud).

The pressure exchange phase, which is usually very short, is followed by the slower transport processes of mixing and heat transfer, and the extent to which these are completed depends, of course, on the length of the interaction duct. The two flows may also be separated from one another after pressure exchange, before their mixing has progressed too far. Since both flows occupy fixed positions in F_s, one obvious way of separating them is to extract them from the interaction space through ports suitably arranged in fixed positions in this frame of reference. Another way is suggested by the already noted difference in orientation of the particle velocities of the two flows in F_u after the deflection phase. If the enclosure of the interaction space is provided with a set of stationary exit passages whose orientation is that of the direction of motion of the primary fluid particles, and with a second set of passages whose orientation is that of the direction of motion of the secondary fluid particles, a predominant portion of each of the two fluids will flow out through that set of passages which matches the orientation of its motion.

The possibility of separating the two flows immediately after pressure exchange is of special interest in applications utilizing the favorable mismatching of total or stagnation pressures that can be made to occur across interfaces in pressure exchange. This effect is not quite as generally achievable in the rotary jet as in the wave rotor pressure exchanger. Indeed, in the latter device, pressure and velocity being both continuous across the interfaces, all that is required for a favorable mismatching is that the density be higher -- or the speed of sound be lower -- on the driven than on the driving side of the interface. In the rotary jet, the velocities are not continuous across the interfaces, but favorable interface mismatchings can nevertheless be generated over a wide range of conditions. Evidence of this effect is provided by the fact that, as will be seen below, the performance of rotary-jet ejectors benefits, above a certain spin angle, from an increase of the secondary-to-primary density ratio, whereas that of the conventional (steady-flow) ejector is always affected adversely by such an increase.

Effective stagnation-pressure mismatching and separation of the two flows after pressure exchange may conceivably make possible the self-sustaining operation of rotary jets (in looped arrangements with combustion chambers) as bladeless versions of gas generators or turbojets [11,24]. In such applications, the amplitude of the mismatching will have to be great enough to cause the stagnation pressure of the secondary flow after pressure exchange to be higher than that of the primary flow before pressure exchange.

The only drawback of the rotary jet relative to the steady-flow ejector is that it has one moving part -- the rotor. Theory [25] suggests, however, that flow patterns similar to those of the primary streams in the rotary jet should be achievable also without moving parts, through the promotion and utilization of stall propagation in stationary cascades of high solidity.

Because of the dominant role of cryptosteady pressure exchange in the first phase of the interaction, the potential energy transfer efficiency and overall performance of the rotary jet can be expected to be considerably better than those of other modes of direct flow induction under the same boundary conditions. This fact has been confirmed experimentally by various researchers [18, 21, 26, 27, 28], although minor changes in design are still occasionally found to result in significant and as yet unexplained changes in

[6] More than that, they are merely patterns and not bodies of abiding material -- a fact that also frees them from temperature and stress limitations and makes them immune to erosion and to damage from cavitation, scraping, and other sources.

performance, indicating that much remains to be done in the area of rotary-jet research.

Existing theories of the rotary jet have focused on its performance as a thrust augmenter and have been based on three main analytical models -- a two dimensional, a "thin-jet-strip" and a "wide-jet-strip" model.

In the two-dimensional model [11, 17, 18] the penetration of the secondary flow into the space between the pseudoblades is assumed to be completed before the two flows deflect each other to a common orientation in the rotor-fixed frame of reference, and the depth of the interaction space is assumed to be small compared to i⁺ mean radius.

In Ho nemser's thin-jet strip approach [18, 19, 20] the primary is treated as a very thin jet successively interacting with infinitesimal layers of the secondary flow. In each of these infinitesimal steps, as the two interacting flows deflect each other to a common orientation, the primary jet, which is finite, undergoes an infinitesimal deflection, and the secondary layer, which is infinitesimal, undergoes a finite deflection. The changes of angular momentum of the two flows in each step must be equal and opposite. The equation expressing this fact yields the distribution of deflections and velocities at the exit from the interaction space.

A more detailed analytical model has been developed by Costopoulos [23], whereby account is taken of that part of the interaction that takes place where the secondary flow enters the space between the pseudoblades. As Fig. 9 shows, different layers in both flows undergo

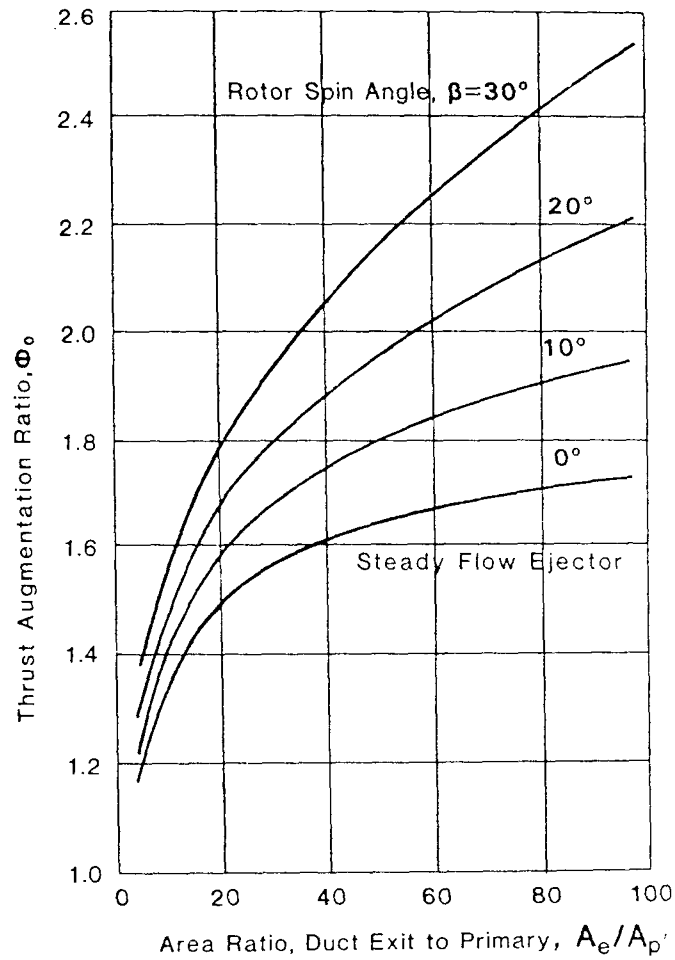

Fig. 10 - Theoretical predictions of thrust augmentation for constant area ducts.

different histories, different deflections and different exchanges of mechanical energy. The performance calculated by this method is plotted in Fig. 10.

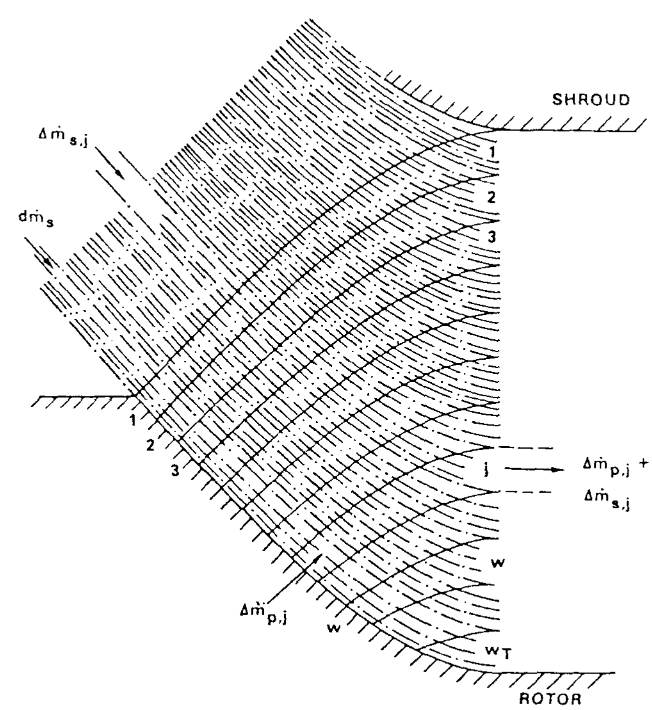

Fig. 9 - Mutual deflection phase according to Costopoulos model [23].

The effect of mixing *during* deflection has been examined by Hohenemser and Porter [20] and by Costopoulos [23] on the basis of the thin-jet strip and of the wide-jet strip model, respectively. This effect is always an adverse one, as one would expect, since energy that is transferred through mixing during the deflection phase is energy that could have been transferred more efficiently through the work of the interface pressure forces. In contrast, mixing *after* the mutual deflection phase is always found to be beneficial, if no account is taken of the drag and weight penalties that are associated with the required extension of the shroud [23]. Actually beyond a certain spin angle (about 15° or 20°), the benefit that can be derived from mixing becomes too small to offset these penalties.

Changing the secondary-to-primary density ratio can have an important effect on performance [17, 18, 23], but the nature of the effect depends on the spin angle. Fig. 11 illustrates this dependence for the case of a

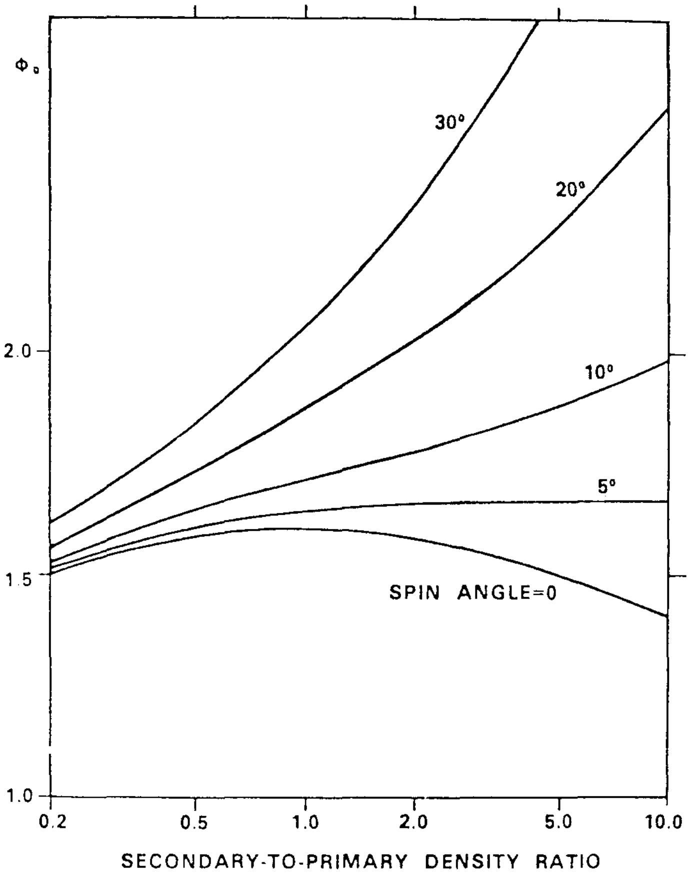

Fig. 11 - Effect of secondary-to-primary density ratio for constant area interactions when Ae/Ap = 30.

constant-area thrust augmenting ejector. The effect of increasing the density ratio above 1.0 is an adverse one for the steady-flow ejector and for the rotary jets of low spin angles, but is otherwise a beneficial one.

Typical results of recently conducted tests under a George Washington University contract with the Air Force Office of Scientific Research are summarized in Table I, in which the performance of a steady-flow (zero spin angle) three-nozzle thrust augmenting ejector is compared with that of a rotary jet of the same shape and size, operating under exactly the same conditions. In this table, L/D_e is the ratio of the length of the shroud to its exit diameter; η_T is the efficiency of energy transfer, i.e., the ratio between the availability gained by the secondary and that lost by the primary in the interaction; and the meaning of the other symbols is as defined under "NOTATION". The diffuser area ratio was 2.36 in all cases. The last three quantities are calculated from the measured thrust augmentation ratio on the assumption of a uniform flow at the augmenter's exit.

It will be noted that even a moderate spin angle, hence a relatively slow rotation, can produce a significant improvement in performance with greatly reduced interaction lengths.

The results indicated in TABLE I were chosen because they demonstrate the advantage of the rotary jet with varying diffuser length, other things remaining constant. However, it should be noted that the thrust augmentations indicated are below those that can be obtained. At The George Washington University thrust augmentations in excess of 1.8 have been obtained and future improvements are expected.

By comparison extensive support has been provided to the steady flow ejector. With the hypermixing nozzle ejector developed at Rockwell [44], typical thrust augmentations of about 1.65 have been obtained.

In Fig. 12, the comparison between rotary jet and steady flow ejectors is further indicated. The ratio of measured thrusts for rotary jets and steady flow ejectors with the same duct, nozzle pressure ratio, and nozzle geometries is plotted vs. duct length to exit diameter ratio for specified rotors and duct diffuser area ratios. Two rotor configurations were used. The data show that the rotary jet provides a major improvement in thrust over the steady flow ejector for relatively short ducts and for high area ratio diffusers.

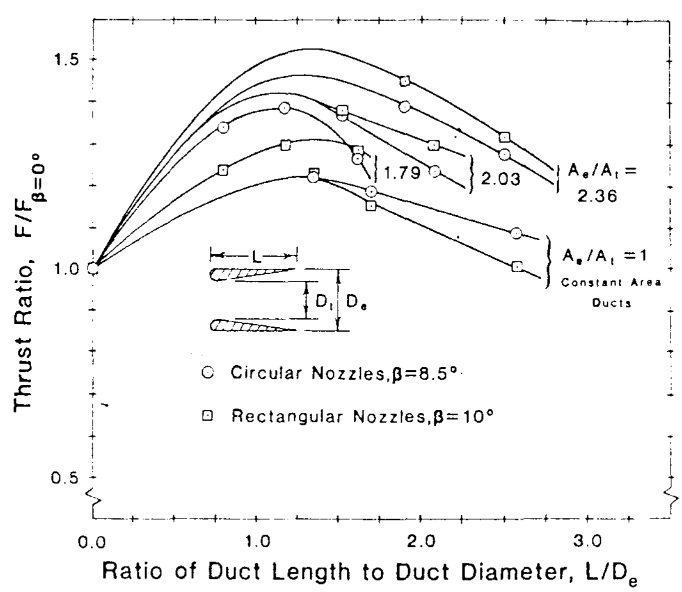

Fig. 12 - Performance comparison of rotary jet and steady-flow ejector

In TABLE II is shown a comparison of theoretical and experimental thrust augmentation and dissipation for constant area ducts. Comparing theoretical and experimental dissipation, δ, it can be seen that in the models tested the dissipation was considerably higher than the theoretical values. Hence, there is much room for improvement in rotary jet performance.

THE CRYPTOSTEADY-FLOW ENERGY SEPARATOR

The only known steady-flow method of energy separation is that of the Ranque-Hilsch tube [29, 30], in which the transfer of energy

TABLE I - Rotary Jet and Steady Flow Ejector Performance with Diffusers of Area Ratio 2.36.

	β (degrees)	Φ_0	δ	μ	η_τ
$A_e/A_p = 46$ $L/D_e = 1.90$ Round Nozzles	0 8.5	1.24 1.64	.796 .690	6.55 7.69	0.181 0.284
$A_e/A_p = 46$ $L/D_e = 2.50$ Round Nozzles	0 8.5	1.41 1.68	.753 .679	7.05 7.79	0.223 0.295
$A_e/A_p = 151$ $L/D_e = 1.90$ Rectangular Nozzles	0 10	1.08 1.61	.909 .834	11.77 14.59	0.084 0.157
$A_e/A_p = 151$ $L/D_e = 2.50$ Rectangular Nozzles	0 10	1.23 1.67	.888 .824	12.63 14.88	.104 .166

TABLE II - Rotary Jet Performance with Constant Area Ducts; Comparison of Theory* and Experiment

A_e/A_p	β	Φ_0 Exp.	Φ_0 Theory	δ Exp.	δ Theory	μ	η_τ	Ref.
16	7	1.36	1.56	.60	.51	3.66	.34	[45]
16	18	1.44	1.75	.57	.42	3.78	.38	[19]
16	24	1.51	1.87	.54	.36	3.91	.41	[19]
40	13	1.80	2.02	.62	.55	7.48	.35	[18]
66	16	1.86	2.46	.69	.53	10.10	.29	[18]
20	45	2.00	2.45	.37	.14	5.32	.59	[22]
120	15	1.56	3.07	.82	.51	12.68	.17	[28]

* Theoretical predictions based on results of [23].

is effected entirely through the dissipative action of viscous stresses and irreversible transport processes, hence with very low efficiency. In an effort to achieve the same effect more efficiently through the utilization of nonsteady interactions, some attention has been given to the development of devices which operate on the basis of wave processes [9,10]. However, the sensitivity of performance to departures from the appropriate mode and timing of control operations (like the opening the closing of ports) represents a serious impediment to the successful design and operation of such devices. In contrast, as has been pointed out earlier, an important advantage of cryptosteady-flow energy separation is that it can be generated, controlled, and analyzed as a steady-flow process in the unique frame of reference in which it is steady (F_s), while retaining all the potential advantages of nonsteady-flow energy exchanges in the frame of reference F_u in which it is utilized.

Consider again the situation described by Fig. 4. A plane and initially uniform jet of an inviscid fluid issues through a nozzle and impinges on an adiabatic wall surface W at an angle θ to the normal to the wall. As already notes, the impingement causes the jet to divide into two subflows, c and h. If the discharge pressures on the c and h sides are equal, the ratio of the mass flow rates $\dot{m}_h$ and $\dot{m}_c$ is $\mu=(1-\sin\theta)/(1+\sin\theta)$.

For either subflow, $\vec{u}=\vec{c}+\vec{V}$, hence

$h^0=h^*+\frac{1}{2}V^2+\vec{c}\cdot\vec{V}$. Since $h_h^*=h_c^*$, there follows

$h_h^0-h_c^0=(\vec{c}_h-\vec{c}_c)\cdot\vec{V}$.

This quantity vanishes only if $\vec{V}$ is either zero or normal to the deflected streams. Excluding these two trivial cases, it must be concluded that energy is transferred, in F_u, from one portion to the other of the original stream as the two portions are deflected to different orientations. The energy transferred is, of course, equal to the work done by the

pressure forces that the two flows are exerting on one another at their contact surface s, Flows c and h can be captured separately in F_u as a "cold" and a "hot" output, respectively.

The plane-flow situation just described may be approximated through lateral confinement of the flow field by means of end plates or vanes. If an array of jets is used, the vanes may be shaped to lead flows c and h into separate spaces (Fig. 13). For the flow to be truly cryptosteady, the vanes must be stationary in F_s.

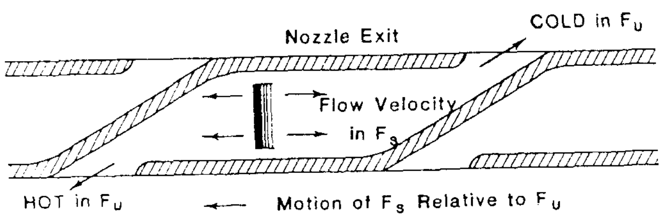

Fig. 13 - Example of output collection in external separation.

The arrangement of Fig. 6(a) approximates that of Fig. 4 so long as the radial depth of the annular space separating the rotor from the casing is small compared to its mean radius. In the arrangement of Fig. 6(b) separation of flows c and h takes place inside the rotor, and the two flows are discharged, through separate nozzles (of which only two are shown), into separate collectors

As mentioned earlier, Figs. 6(a) and 6(b) are schematic representations of "external separation" and "internal separation" arrangements respectively.

Cryptosteady-flow energy separation was first proposed and analyzed in reference [31]. More comprehensive studies of the concept are presented in references [32] through [41].

The validity of the concept was first tested in experiments on small and rather crude models at Rensselaer Polytechnic Institute [31]. With a hot-to-cold mass flow ratio (μ) of 0.5 and an input flow temperature of 70°F (20°C), the measured cold-out-put temperature was found to decrease from 48°F (+8.9°C) to 30°F (-1.1°C) as the pressure ratio (ratio of inlet stagnation to discharge static pressure) was increased from 2.0 to 4.0. Neglecting heat losses and bearing friction or other shaft torque, the stagnation enthalpy rise on the hot side is, of course, always equal to the stagnation enthalpy drop on the cold side divided by the mass flow ratio . Thus, when the cold-output temperature (T_c^0) was 48°F (+8.9°C), the hot-output temperature (T_h^0) was 94°F (34.4°C); and when T_c^0 was 30°F (-1.1°C), T_h^0 was 110°F (43.3°C).

The high promise of this mechanism was confirmed by experiments conducted on an R.P.I. model at Columbia Research Corp. [42], where, for example, with a mass flow ratio of 0.3, a pressure ratio of 3.0 and an input flow temperature of 54.5°F (12.5°C), the measured temperature of the hot output was 148.2°F (64.5°C) and that of the cold output was -11.2°F (-24°C).

Experimental work on both internal-and external-separation devices has only recently been resumed, under the sponsorship of the U.S. Department of Energy, at The George Washington University [43].

In general, test results so far have been found to be in varying degrees of agreement -- from good to poor, depending on the model -- with the predictions of the available theories. Where the agreement has been unsatisfactory, the discrepancy has helped in some cases to identify design deficiencies and/or needed refinements of the theory. For example, in the case of an internal-separation model, the nozzle efficiencies were found to be only about 65%, far lower than the nozzle efficiencies that can be achieved through appropriate design. Collection losses have also been found to be high, particularly on the hot side. These and other observations indicate that there is a great deal of room left for improvement in energy separator design.

The most comprehensive existing analysis of energy separation in all its forms, including in particular the cryptosteady modes, is to be found in P.A. Graham's doctoral dissertation [33]. The performance chart of Fig. 14 is taken from this work. Here u_0 is the particle velocity on isentropic expansion from inlet conditions to discharge pressure. The baseline configuration is an "internal" one in which nozzle inclinations and discharge pressures are equal on the two sides and there is no prerotation.

In the study of external separation, special attention has been given to the loss of performance that may result from the boundary layer entrainment of hot gas to the cold-output side [35]. The prediction of the theory, when applied to representative configurations and operating conditions, has been that the mass entrainment should not increase the cold-side stagnation enthalpy by more than a fraction of a percent, and this prediction has been confirmed by simple experiments.

In the study of internal separation, attention has been given to the effects of asymmetries in nozzle efficiencies, prerotation velocities, back pressures and other variables, and also to the nonsteady-flow effects resulting from the motion of the rotor discharge nozzles relative to the supply channels; and criteria have been developed for the identification of the conditions for maximum separation of energy under any given set of constraints [34, 36, 37, 41].

The potential performance of the energy separator has been assessed in various areas of applications. In two studies [28, 39], an evaluation has been made of the performance of the energy separator in home and vehicular air

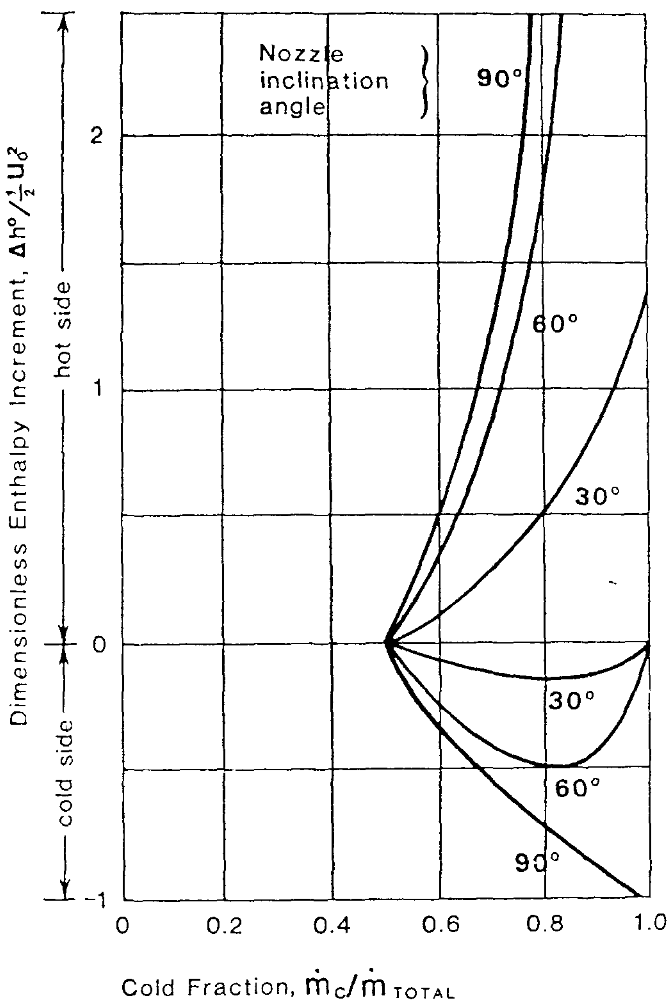

Fig. 14 - Effect of nozzle inclination on hot-side and cold-side dimensionless enthalpy increments for a baseline internal energy separator [33].

conditioning, with various modes of energy recovery on the hot side. The main conclusion is that in such applications the coefficient of performance of the energy separator can be lower or higher - even very much higher - than that of conventional air cycle refrigerators, air conditioners, and heaters, depending on the extent to which the mechanical energy available in the hot output is recovered through a turbine or otherwise (e.g., through an absorption cycle or a Rankine power cycle).

Another assessment [40] has dealt with the limits of performance of the "basic" device, i.e., of the energy separator alone, without any of the machinery that may be required for the recovery of any portion of the available mechanical energy of the hot output. Three areas of application have been considered in this study: (1) as a heat pump; (2) in the concurrent heating and cooling of separate spaces, and (3) in the air-conditioning and cooling of high-speed vehicles, where the boundary-layer temperature is too high to make it practicable to exchange heat with the external flow. The results have confirmed that the energy separator has very attractive performance capabilities in all three areas. In the latter application, there is an advantage in just dumping the hot output overboard, with or without utilization of its momentum for thrust, thereby eliminating the need for a heat exchanger.

Cryptosteady-flow energy separation could probably be used to advantage also in truck and railroad car refrigeration, on-board-aircraft oxygen generating devices, and perhaps also in cryogenics, desalination, and the chemical process industry.

The cryptosteady-flow energy separator erives its potential merit primarily from ie fact that it permits the energy extracted rom the cooling subflow to appear in the other subflow not as heat but rather in the form of recoverable mechanical energy. Achievement of this potential will, of course, be contingent on the successful resolution of still outstanding problems of hot-output collection. There are, however, applications -- such as the above - mentioned air conditioning and structural cooling of high-speed aircraft -- in which energy separation can provide important advantages over conventional systems, even in the absence of mechanical energy recovery.

ACKNOWLEDGEMENT

This work was supported by the U.S.A.F. Office of Scientific Research under Contract No. AFOSR-F49620-80-C-0043 and by the U.S. Department of Energy under Contract No. DE-AS05-83ER13142.

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"INTERNAL" MODES OF CRYPTOSTEADY-FLOW ENERGY SEPARATION

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ABSTRACT

Previous analyses of fluid dynamic energy separation are extended to account for all pertinent variables and asymmetries, with focus on the cryptosteady internal-separation modes. Results are presented in a way to facilitate the identification of the conditions for maximum separation of energy under any given set of constraints.

NOTATION

- A total cross-sectional area of discharge nozzles on one side
- c fluid particle velocity relative to rotor
- C $= c/u_o$
- F $= L/r_{dc}$
- F_s = frame of reference fixed to rotor
- F_u = laboratory frame of reference
- h = specific static enthalpy
- h^o = specific stagnation enthalpy in F_u
- h^* = specific stagnation enthalpy in F_s
- h_{ds} $= h_d^*(p_d/p_d^*)^{\frac{\gamma-1}{\gamma}}$
- H $= h^o/\frac{1}{2}u_o^2$
- ΔH_c $= (h_i^o - h_{dc}^o)/\frac{1}{2}u_o^2$
- ΔH_h $= (h_{dh}^o - h_i^o)/\frac{1}{2}u_o^2$
- L externally-applied torque (positive if a driving torque)
- $\dot{m}$ mass flow rate
- M prerotation momentum per unit mass
- p static pressure
- p_o reference pressure (normally ambient pressure)
- p^o stagnation pressure in F_u
- p^* stagnation pressure in F_s
- P $= (p/p_o)^{\frac{\gamma-1}{\gamma}}$; $P^o = (p^o/p_o)^{\frac{\gamma-1}{\gamma}}$
- r distance from rotor axis
- u fluid particle velocity in F_u
- u_o $= [2h_i^o(1 - \frac{1}{P_i^o})]^{\frac{1}{2}}$ particle velocity on isentropic expansion from (h_i^o, p_i^o) to p_o.
- u' prerotation velocity (defined as M/r_d)
- U' $= u'/u_o$
- v $= \omega r$
- V $= v/u_o$
- γ ratio of specific heats
- δ = cos θ (negative on "cold" side)
- η $= \dfrac{h_d^* - h_d}{h_d^* - h_{ds}}$ (nozzle efficiency)
- θ angle $(\vec{v}, \vec{c})$
- μ $= \dot{m}_h/\dot{m}_c$
- ρ density
- τ $= r_{dh}/r_{dc}$ (radius ratio)
- ω angular velocity of rotor

Subscripts

- c "cold" discharge side
- d rotor nozzle discharge
- h "hot" discharge side
- i energy separator input flow

INTRODUCTION

The fluid dynamic energy separator (hereinafter to be referred to simply as the "energy separator") is a device which divides an originally homogeneous flow into two subflows, in one of which the availability is increased at the expense of the availability in the other.

The energy separator can take a variety of forms, but basically its function is in all cases that of the back-to-back turbine-compressor arrangement shown in Fig. 1, where c and h denote the "cold" (de-energized) and the "hot" (energized) subflow, respectively. The two outputs may be used for the simultaneous cooling and heating of separate spaces, or the mechanical energy available in the hot output may be recovered in any of a variety of ways; and there are applications in which energy separation can serve a useful purpose even when one of the two outputs is not utilized at all.[1]

From the perspective of the second law of thermodynamics, the energy separator derives its potential merit from the fact that some if not all of the energy it extracts from one subflow appears in the other subflow not as heat but rather in its highest-quality form, as available mechanical energy. Because of this feature, energy separation can offer important advantages over competing mechanisms in a number of applications.

The only known steady-flow method of energy separation is that of the "vortex tube", first proposed and experimentally demonstrated by Ranque [1][2] and later systematically tested by Hilsch [2]. In the vortex tube, the transfer of energy from one subflow to the other takes place primarily through the work of viscous forces [3], hence at a considerable loss of availability. The transfer can be effected more efficiently through the essentially nondissipative work of interface pressure forces ("pressure exchange"), but this requires that the interaction be nonsteady, because for such forces to do work the interfaces must move.

One obvious way of generating the required nonsteadiness is through valve action, as is done in "pressure-exchanger dividers" [4,5], which are energy separators operating on the basis of wave processes. Valving, however, generates throttling and other flow losses, and the operation of these devices entails, in addition, the usual limitations and control difficulties of valve machines.

Penalties of this sort are for the most part absent in energy separators utilizing cryptosteady interactions, i.e., interactions which, while nonsteady in the frame of reference F_u in which they are utilized, admit a unique frame of reference F_s in which they are steady [6,7]. Dissipation, being an invariant with respect to changes of the frame of reference, is the same in F_u as in F_s. It follows that no dissipation is inherently involved in the generation of pressure exchange in cryptosteady energy separation. The change of frame of reference is most conveniently achieved in practice by rotating F_s relative to F_u: frame F_u is the laboratory frame of reference, and frame F_s is fixed to a rotor into which the working fluid is fed under pressure and out of which it is discharged through peripheral nozzles [7 through 11]. Most studies of this process so far have dealt only with free-spinning rotors, driven by the resultant reaction of the issuing jets, but this is not a necessary constraint. Rotation of the rotor inevitably involves windage and bearing-friction losses, but these can be made small and often negligible by appropriate design.

Cryptosteady-flow energy separators may be classified according to whether the separation of the two subflows from one another takes place outside or inside the rotor (see Figs. 2 and 3). More detailed descriptions of the two-classes have been presented in previous papers [7 through 12]. The potential merits of cryptosteady-flow energy separation have been assessed by Hashem in a comparison of steady and cryptosteady modes [8] and by Graham in a comprehensive study of all three modes (steady, nonsteady, and cryptosteady) [9], as well as in subsequent analyses [10,11, 12].

The present paper focuses on the internal-separation modes, but extends previous analyses of these modes to cover all pertinent variables, and it does so with a view to facilitating the identification of those arrangements that will optimize performance in any specific application, under any given set of boundary conditions.

ANALYSIS

Figure 4 describes the analytical model that is to be considered here. The working fluid enters the energy separator at i, and a portion of it is led to the "cold output" nozzles c, while the remainder is led to the "hot output" nozzles h. Stationary vanes or baffles impart prerotation to the subflows. As will be seen below, there are advantages to be gained from the combined effect of a positive prerotation on the cold side and a negative one on the hot side. In the arrangement of Fig. 4, this is accomplished by a single, helical baffle. The orientation δ of the discharge nozzles is positive on the hot side and negative on the cold side, so that the cold output exerts on the rotor a positive torque and the hot output a negative one; and the nozzles are so dimensioned and arranged that the former overcomes the latter [7,10]. Thus, in flowing through the rotor, subflow c is de-energized and subflow h is energized.

The following analysis accounts for all pertinent variables and for possible asymmetries of their values on the two sides -- differences of nozzle area, orientation, and discharge radius, of nozzle efficiency, of prerotation velocity, and of discharge pressure -- and also for the effect of an externally applied torque.

The following assumptions are made:

1. The working fluid is an ideal gas with constant specific heats.
2. The prerotation flows are irrotational.
3. The two subflows are independently adiabatic within the energy separator.

The angular momentum of the prerotation flow, M, and the externally applied torque, L, are taken to be positive when in the direction of rotation of the rotor.

The relation

$$h_d^* = h_i^o + \tfrac{1}{2}v_d^2 - M\omega$$
$$= h_i^o + \tfrac{1}{2}v_d^2 - u'v_d \qquad (1)$$

---

[1] A case in point is that of the air conditioning and structural cooling of high-speed aircraft, where the high boundary-layer air temperature makes it impracticable to transfer heat to the surroundings through heat exchangers. The energy separator can dispose of the extracted energy by dumping the hot output overboard, with or without utilization of this output's momentum for thrust.

[2] Numbers in brackets designate References at end of paper.

In nondimensional form, and remembering that $\delta_h > 0$ and $\delta_c < 0$, Eqs. (2) and (6) become

$$C_{dc}^2 = \eta_c \left[V_{dc}(V_{dc}-2U_c') + H_i\left(1-\frac{p_{dc}}{p_i^o}\right)\right]$$

$$C_{dh}^2 = \eta_h \left[\tau V_{dc}(\tau V_{dc}-2U_h') + H_i\left(1-\frac{p_{dh}}{p_i^o}\right)\right] \qquad (9)$$

$$\mu = \frac{1}{\tau}\,\frac{C_{dc}\,|\delta_c| - V_{dc} + U_c' + F/\dot{m}_i u_o}{C_{dh}\,|\delta_h| + \tau V_{dc} - U_h' - (1/\tau)F/\dot{m}_i u_o} \qquad (10)$$

Finally, Eqs. (7), (8), and (10) yield

$$\frac{A_h}{A_c} = \mu\,\frac{p_{dc}C_{dc}}{p_{dh}C_{dh}}\,\frac{H_i+(\tau V_{dc}-2U_h')\tau V_{dc}-C_{dh}^2}{H_i+(V_{dc}-2U_c')V_{dc}-C_{dc}^2} \qquad (11)$$

If $\eta_c=\eta_h=\tau=|\delta|=P_{dc}=P_{dh}=1.0$ and $u_c'=u_h'=L=0$, Eqs. (9), (10), and (11) reduce to

$$C_d^2 = V_d^2 + 1$$

and

$$\frac{A_h}{A_c} = \mu = \frac{C_d - V_d}{C_d + V_d}$$

Therefore,

$$V_d = \frac{1-\mu}{2\sqrt{\mu}} \qquad (12)$$

or

$$V_d = \frac{1 - A_h/A_c}{2\sqrt{A_h/A_c}} \qquad (13)$$

Finally, Eq. (5) gives, for this condition,

$$\Delta H_c = 1 - \mu$$
$$\Delta H_h = (1-\mu)/\mu \qquad (14)$$

The same condition, but with $|\delta| = 0.966$ on both sides ($\theta=15^o$, $\theta_c=165^o$) and without specification of the value of τ, will be taken in the following as a baseline.

RESULTS AND CONCLUSIONS

Typical results of the analysis developed in the preceding section are presented in Figs. 5 through 12. The mass flow ratio μ is normally specified in advance, being selected on the basis of the use that is to be made of the two outputs. The measure of performance is taken to be the drop of nondimensional specific stagnation enthalpy on the cold side, $\Delta H_c=(h_i^o-h_{dc}^o)/\frac{1}{2}u_o^2$. The stagnation enthalpy rise on the hot side, $\Delta H_h=(h_{dh}^o-h_i^o)/\frac{1}{2}u_o^2$, is readily calculated as $\Delta H_h=\Delta H_c/\mu$, except where energy is exchanged with the surroundings through the work of an applied torque. The rotor speed is represented by the ratio $V_{dc}=v_{dc}/u_o$.

applies to each subflow [10].

From Eq. (1) and the definition of η, there follows

$$c_d^2 = 2\eta\left[\tfrac{1}{2}v_d^2 - u'v_d + h_i^o\left(1 - \frac{p_d}{p_i}\right)\right] \qquad (2)$$

Also,

$$u_d^2 = c_d^2 + v_d^2 + 2c_d v_d \delta \qquad (3)$$

and

$$h_d^o - h_d^* = \tfrac{1}{2}(u_d^2 - c_d^2) \qquad (4)$$

Eqs. (1) through (4) yield, for each of the two stagnation enthalpy increments in F_u,

$$h_d^o - h_i^o = v_d^2 + c_d v_d \delta - u'v_d \qquad (5)$$

and these two increments are related to one another and to the work of the externally applied torque through the energy equation

$$\mu = \frac{h_i^o - h_{dc}^o + \frac{L\omega}{\dot{m}_i}}{h_{dh}^o - h_i^o - \frac{L\omega}{\dot{m}_i}} \qquad (6)$$

On the other hand,

$$\mu = \frac{A_h\, c_{dh}\, \rho_{dh}}{A_c\, c_{dc}\, \rho_{dc}} \qquad (7)$$

and, if both flows are discharged fully expanded to their respective back pressures p_{dc} and p_{dh},

$$\frac{\rho_{dh}}{\rho_{dc}} = \frac{p_{dh}}{p_{dc}}\,\frac{h_{dc}}{h_{dh}}$$
$$= \frac{p_{dh}}{p_{dc}}\left[\frac{h_{dc}^* - \frac{1}{2}c_{dc}^2}{h_{dh}^* - \frac{1}{2}c_{dh}^2}\right] \qquad (8)$$

The area ratio A_h/A_c can be calculated from Eqs. (6), (7), and (8).

The discharge static enthalpies are both uniquely determined once the input stagnation state, the discharge static pressures, and the nozzle efficiencies are specified. The highest ΔH_c is, therefore, obtained under the conditions which minimize u_{dc}.

Unless otherwise indicated, the results presented here are independent of the pressure ratio p_i^o/p_o.

Figure 5 shows the effects of changes of the radius ratio τ and of the mass flow ratio μ on ΔH_c, V_{dc}, and A_h/A_c, for baseline configurations. To each value of τ there corresponds an "optimum" value of μ (one that maximizes ΔH_c), and to each value of μ an optimum value of τ, but $(\Delta H_c)_{max}$ itself is relatively insensitive to such changes, and so is V_c in the optimum condition. Departures from the baseline conditions will, of course, result in modifications of these results (see, e.g., Fig. 11).

Figures 6 through 12 show the effects of individual departures of certain variables from their baseline values. The mass flow ratio μ is chosen to be 0.2 in all cases.

Figure 6 reveals that, all else being equal, a change of nozzle efficiency on the cold side has a considerably larger effect on performance than an equal change on the hot side. For example, with a fixed $\eta_h = 0.9$, a decrease of η_c from 0.9 to 0.7 causes $(\Delta H_c)_{max}$ to decrease by over 30%, whereas a decrease of η_h from 0.9 to 0.7 with η_c fixed at 0.9 has almost no discernible effect on $(\Delta H_c)_{max}$. The peripheral speed of the rotor is also more sensitive to changes of nozzle efficiency on the cold than on the hot side. (These effects are actually of opposite signs, V_{dc} increasing or decreasing depending on whether η is increased on the cold or on the hot side. Changes of the pressure ratio p_i^o/p_o have no effect on either ΔH_c or V_{dc} but do modify the required area ratio A_h/A_c.

As already noted, some benefit can be derived from dual prerotation (positive on the cold side and negative on the hot side). This can be seen in Fig. 7. Again, the dominant effect is produced by changes on the cold side. The negative prerotation on the hot side has a very weak effect on performance (and not always a beneficial one at that), but it is useful nevertheless in that it has a moderating effect on the rotor speed for any given ΔH_c.

If the discharge pressures p_{dc} and p_{dh} are equal and their common value is taken as the reference pressure p_o, changes of the input pressure ratio p_i^o/p_o, all else being the same, have no effect on the nondimensional quantities ΔH_c, V_{dc}, and A_h/A_c. On the other hand, the effect of asymmetries in the back pressures is strongly dependent on the input pressure ratio. Figs. 8 and 9 show such effects for $p_i^o/p_o = 1.136$ and $p_i^o/p_o = 2.0$, respectively. Changes on the cold side have, here too, a far larger effect on performance than comparable changes on the hot side. It will be noted that the largest ΔH_c is obtained in all cases with a radius ratio τ lower than 1.0, but a τ somewhat larger than this optimum makes possible an important reduction in rotor speed at an insignificant sacrifice of performance.

The effect of changes of the input pressure ratio when the discharge pressures are unequal is further illustrated in Fig. 10 for the case of an energy separator with fixed and strongly asymmetrical back pressures $P_h = 1.1$ (i.e., $p_{dh}/p_o = 1.396$) and $P_c = 0.5$ (i.e., $p_{dc}/p_o = 0.088$).

Whereas Fig. 5 compares the merits of different values of τ and μ for a fixed $|\delta|=0.966$, Fig. 11 compares the merits of differents values of τ and $|\delta|$ for a fixed mass flow ratio (again $\mu=0.2$). Curves are plotted for $|\delta|=1.0$ ($\theta_h=0^o$, $\theta_c=180^o$), $|\delta|=0.966$ ($\theta_h=15^o$, $\theta_c=165^o$), $|\delta|=0.866$ ($\theta_h=30^o$, $\theta_c=150^o$), and $|\delta|=0.707$ ($\theta_h=45^o$, $\theta_c=135^o$). As could be expected, $|\delta|=1.0$ produces the best performance, but also the highest rotor speed. It will also be noted that to each nozzle inclination there corresponds an optimum value of τ.

Finally, the effect of bearing friction is shown in Fig. 12 as an illustration of the effect of an externally-applied torque. The resisting torque due to axial and radial loads is assumed to be negligible, and the lube torque is assumed for simplicity to be linearly proportional to the angular velocity, $L = -K\omega$. Then, $F/\dot{m}_i u_o = -KV_{dc}/\dot{m}_i r_{dc}^2$. In the example considered here, the rotor shaft was supported by two Barden 101 H precision bearings, and it was found that over a selected range of rotor speeds the value of K could, in fair approximation, be taken to be 2.17×10^{-5} N-m-s or kg·m²/s (5.12×10^{-4} lbm·ft²/s). Two cases are considered: (1) $\dot{m}_i = 1.1325$ kg/s (2.5 lbm/s), $r_{dc} = 0.3048$m (1.0 ft); and (2) $\dot{m}_i = 0.0907$ kg/s (0.2 lbm/s), $r_{dc} = 0.0762$m (0.25 ft). Curves 1 and 2 in Fig. 12 refer to cases 1 and 2, respectively. The results for case 1 are indistinguishable from those calculated for L=0, i.e., in the absence of bearing friction. On the other hand, the results for case 2 show that the adverse effect of bearing friction can be quite significant, particularly on the hot-output side, for small models and/or at low mass flow rates.

It should, of course, be remembered that the information provided by Figs. 5 through 12 applies in each case only to the effects of changes of individual parameters, and cannot be used, therefore, in any but qualitative estimates of the effects of concurrent changes of various parameters. The charts of Figs. 5 through 12 are presented here primarily to show trends and to illustrate the manner in which the analysis developed in the preceding section can be applied toward the evaluation of the attainable performance under specified constraints, and in the determination of the design and operational variables that will best meet any given set of specifications.

ACKNOWLEDGMENT

This work was supported by U.S. Department of Energy under Contract No. DE-AS05-83ER13142.

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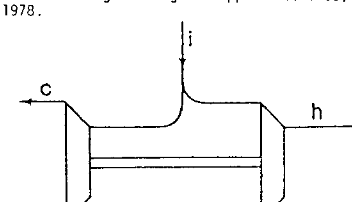

FIG. 1

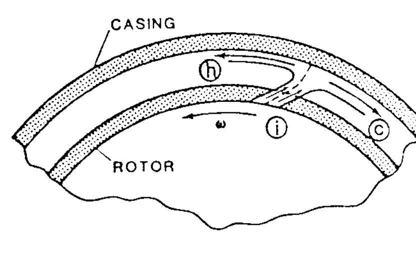

FIG. 2 Schematic of external separation

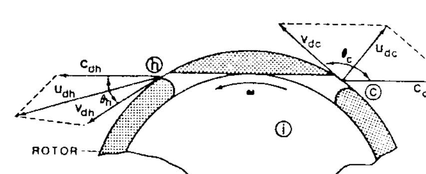

FIG. 3 Schematic of internal separation

FIGURES 6 THROUGH 12 ARE FOR $\mu = 0.2$

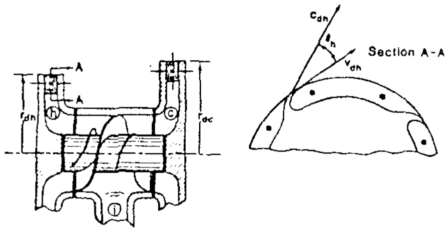

FIG. 4 Schematic of analytical model

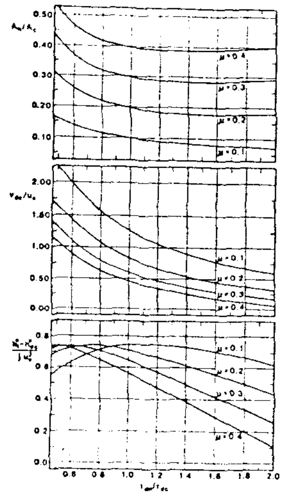

FIG. 5 Results for baseline configurations

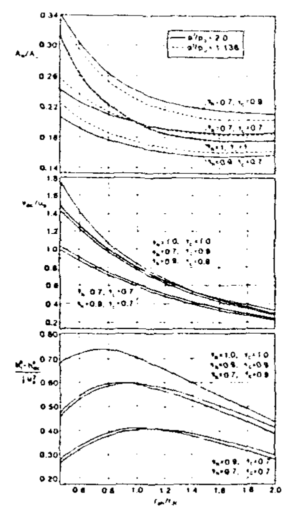

FIG. 6 Effects of nozzle efficiencies

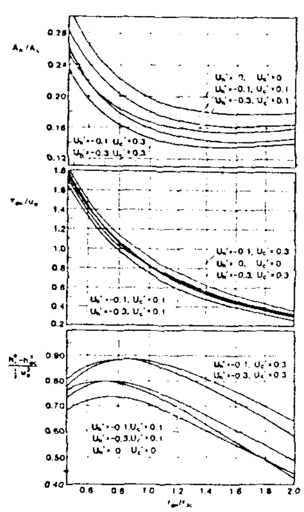

FIG. 7 Effects of asymmetric prerotations

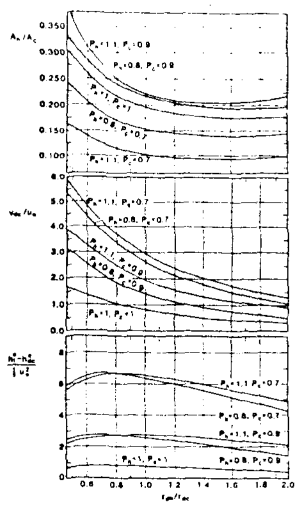

FIG. 8 Effect of differential back pressures when $p_i^0/p_0 = 1.136$

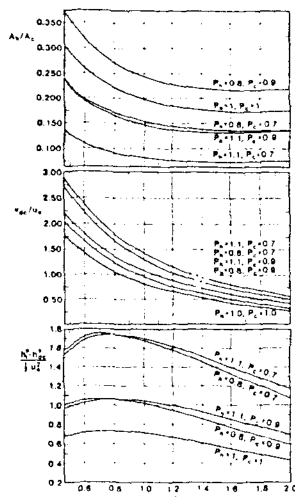

FIG. 9 Effect of differential back pressures when $p_i^0/p_0 = 2.0$

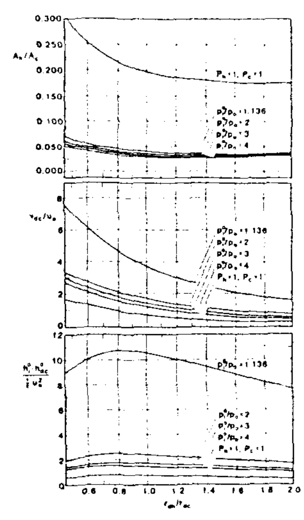

FIG.10 Effects of changes of p_i^0/p_0 when $P_h = 1.1$ and $P_c = 0.5$

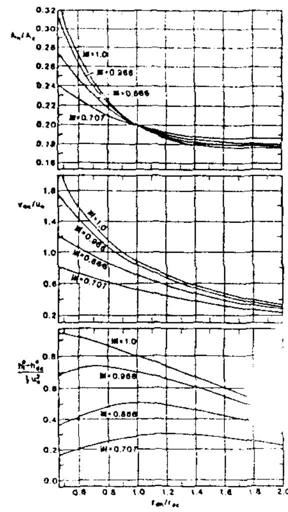

FIG.11 Effect of nozzle inclination

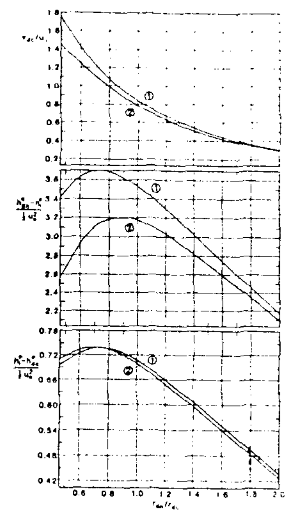

FIG.12 Effects of bearing friction

Curve 1 is for $\dot{m}_i = 2.5$ lbm/s, $r_{dc} = 1.0$ ft

Curve 2 is for $\dot{m}_i = 0.2$ lbm/s, $r_{dc} = 0.25$ ft

Journal of Fluids Engineering/Transactions of the ASME
December 1973, p. 513-518

Cryptosteady-Flow Energy Separation

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The mechanism of cryptosteady-flow energy separation is described and analyzed in its most general form, with full consideration of the effects of bearing friction or other rotor torque, and of such asymmetries as unequal discharge pressures, peripheral velocities, flow losses, prerotation velocities, and discharge angles. Equations are also developed for the proportioning of rotor nozzles in accordance with performance specifications.

Introduction

CRYPTOSTEADY energy separation is a process[1] whereby the total head or total specific enthalpy of a portion of a flow is increased at the expense of the corresponding quantities in the remainder of the same flow, through direct and essentially nondissipative exchange of energy.

It is known that reversible transfers of mechanical energy in flow systems are possible only where the interacting flows are nonsteady [1].[2] Indeed, in the only known steady-flow mechanism of redistribution of energy within an initially homogeneous flow—that of the Ranque-Hilsch tube [2, 3]—the transfer of energy is effected, rather inefficiently, through the action of viscous stresses. On the other hand, considerably better performance has been shown to be possible when the energy transfer is effected by "pressure exchange," i.e., through the work of interface pressure forces; and pressure exchange is always a nonsteady process, because no work is done by pressure forces acting on a stationary interface.

In an effort to improve energy separator performance through the utilization of pressure exchange, some attention has been given during the last decade to devices called "dividers," which operate on the basis of wave processes [4, 5]. There is reason to believe, however, that the development of practical and efficient dividers would be very difficult, because of the sensitivity of these devices to such factors as imperfect timing of moving mechanical parts to wave and flow processes, diffusion of interfaces, noninstantaneous opening and closing of valves, distortion of shock fronts, etc., not to mention the usual analytical complexities of nonsteady-flow processes.

This paper deals with a nonsteady-flow method of energy separation in which the difficulties just mentioned are overcome through the utilization of "cryptosteady" pressure exchange—a cryptosteady process being defined as one that is nonsteady but admits a frame of reference in which it is steady. The special merit of cryptosteady processes is that they can be generated, controlled, and analyzed as steady-flow processes in this unique frame of reference, while retaining all the potential advantages of nonsteady flows in the frame of reference in which they are utilized.

A simple interaction of this type is shown in Fig. 1. Here two flows deflect each other to a common orientation in a frame of reference F_s in which they are both steady. Apart from transport processes, no energy is exchanged between the two flows in this frame of reference. A transfer of energy does, however, take place—by pressure exchange—in the frame of reference F of an observer O moving at an arbitrary velocity V relative to F_s. The energy so transferred is equal to the work done by the pressure forces which the interacting flows exert on one another at their interface. This work is zero in F_s, where the interface is stationary, but not in F, where the interface moves. Since changes of the frame of observation are reversible, these energy exchanges are essentially nondissipative. Note that, because of the existence of a frame of observation in which the flow is steady (frame F_s), the flow in F is cryptosteady. Analyses of cryptosteady interactions and discussions of some of their applications have been presented in previous papers [6, 7, 8, 9, 10].

The operation of the cryptosteady energy separator may be explained in a similar manner [11], through consideration of a simple two-dimensional situation, such as that shown in Fig. 2. Here a plane and initially homogeneous stream i is seen issuing from a nozzle as a jet impinging on a wall W. The flow field is stationary in a frame of reference F_s, which is the coordinate system fixed to the nozzle. Body forces are assumed to be absent, and viscous stresses and heat exchanges with the surroundings are assumed to be negligible.

The impingement causes the jet to divide into two separate streams a and b, interfacing with one another at the stagnation stream surface s. For example, if the discharge pressure is the same on the two sides, in the same frame, the mass flow rates in the two streams is $\mu = (1 - \sin\beta)/(1 + \sin\beta)$.

The specific stagnation enthalpy (or the total head, if the fluid is incompressible) is, in frame F_s, the same in the deflected flows as in the original stream. This, however, is not true in any other frame of reference. In particular, letting c and u denote fluid particle velocities relative to F_s and to the frame of reference F_u of an observer moving relative to F_s at an arbitrary velocity V relative to the wall, respectively, one has u = c + V, hence

$$\frac{1}{2}(u_b^2 - u_a^2) = \frac{1}{2}(c_b^2 - c_a^2) + (c_b - c_a)\cdot V$$

So long as V $\neq$ 0, the term $(c_b - c_a)$.V never vanishes, because c_b and c_a have different orientations. Therefore, $\frac{1}{2}(u_b^2 - u_a^2) \neq \frac{1}{2}(c_b^2 - c_a^2)$, and since the thermodynamic states are invariant with respect to changes of the frame of reference, there follows

$$h_b^0 - h_a^0 \neq h_b^* - h_a^*$$

Thus if $h_b^* = h_a^*$, $h_b^0 \neq h_a^0$. Since the original stream i is seen as a homogeneous stream in every coordinate system, it must be concluded that energy is transferred, in F_u, from one portion to the other of this stream, as these two portions are deflected to different orientations. This fact can also be explained on the basis of the observation that in frame F_u the interface s is moving and the interface pressure forces are therefore doing work. The energy that is transferred from a to b is, of course, the work done by a on b in this pressure exchange interaction.

As pointed out in references [11] and [12], a situation approximating that of Fig. 2 may be obtained, with a stream of finite transverse dimensions, through lateral confinement of the deflection region by means of end plates or vanes. These vanes must be shaped to lead the deflected flows into separate spaces. As a consequence, the flow can be strictly cryptosteady only if the confining vanes are stationary in F_s.

The motion of F_s relative to F_u is most simply maintained by the reaction of the issuing jet itself. In the situation of Fig. 2, the source may be a nozzle which is constrained to move in a direction parallel to the wall W. A more practical arrangement is that of Fig. 3, where jets issuing from slanted nozzles or slots on the surface of a free-spinning rotor impinge on the internal surface of an enshrouding wall. This arrangement approximates that of Fig. 2, so long as the radial depth of the annular impingement-deflection space is small compared to its mean radius.

In contrast to the "external separation" configuration of Fig. 3 (where the separation of the two flows takes place outside the rotor), Fig. 4 shows an "internal separation" arrangement. Here the separation of the two flows takes place inside the rotor. The two flows are discharged through separate nozzles, of which only two are shown. A schematic view of another internal-separation arrangement, defining some of the nomenclature used in this paper, is shown in Fig. 5. In either case, say for simplicity that bearing friction is negligible, the discharge pressure is uniform, the nozzle inclinations to the rotor surface are equal and opposite, and the internal flow losses are the same for both flows. Then, if the nozzle areas are unequal, the rotor will rotate

[1] Note that this definition differs from those used in previous papers on this subject.

[2] Numbers in brackets designate References at end of paper.

Contributed by the Fluids Engineering Division and presented at the Applied Mechanics and Fluids Engineering Conference, Atlanta, Ga., June 20–22, 1973, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Manuscript received at ASME Headquarters, March 21, 1973. Paper No. 73-FE-24.

Nomenclature

c = fluid particle velocity in F_s
F_s = frame of reference in which the flow is steady
F_u = frame of reference in which the energy separation is utilized
h = specific static enthalpy
h^0 = specific stagnation enthalpy in F_u
h^* = specific stagnation enthalpy in F_s
L = externally applied rotor torque (positive if driving torque)
$\dot{m}$ = mass flow rate
M = angular momentum (per unit mass) of input flow
p = static pressure
p^0 = stagnation pressure in F_u
r = distance from rotor axis
u = fluid particle velocity in F_u
u' = "prerotation" velocity (defined as M/r_d)
V = velocity of F_s relative to F_u
V = ωr
α = ratio of total nozzle exit area on b side to total nozzle exit area on a side
β = inclination of nozzle axis to normal to V in external-separation devices (see Fig. 2)
γ = ratio of specific heats
δ = cos θ (negative on a side)
η = $(h^* - h_d)/(h^* - h_o)$ (nozzle efficiency)[1]
θ = angle (V, c)
κ = $\dot{m}_a(h_i^0 - h_a^0)/\dot{m}_i V_a^2$ (cooling capacity coefficient)
λ = p_d^*/p_i^*
μ = $\dot{m}_b/\dot{m}_a$ (mass flow ratio)
ν = $\dot{m}_a/\dot{m}_i$ (cold fraction)
ρ = density
ω = angular velocity of the rotor

Subscripts

a = flow discharged with $\delta < 0$
b = flow discharged with $\delta \geq 0$
d = rotor nozzle discharge
e = rotor nozzle entrance
i = energy separator input flow
o = conditions resulting from isentropic discharge from p_i^0 to p_d. For example,

$$u_{oa}^2 = 2h_i^0\left[1 - (p_{da}/p_i^0)^{\frac{\gamma-1}{\gamma}}\right]$$

Superscripts

^{0} = stagnation quantities in F_u
* = stagnation quantities in F_s

Assumptions

1 The fluid, when compressible, is assumed to be a calorically perfect gas.
2 In external-separation configurations, the radial distance between rotor and stator is small compared to the rotor radius.
3 Heat exchanges with the surroundings, in the energy separator, are negligible.
4 Prerotation velocities are everywhere parallel to V.

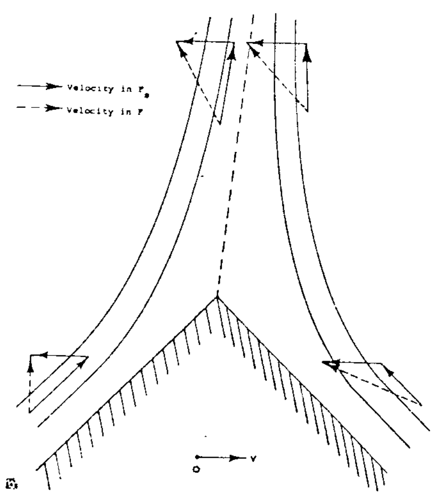


Fig. 1 Schematic of cryptosteady interaction

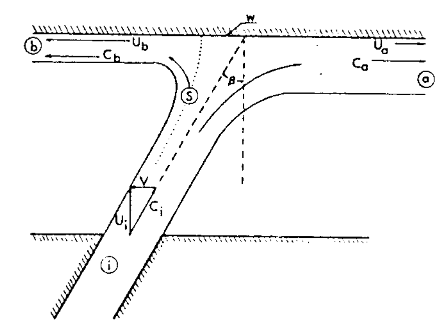


Fig. 2 Schematic of cryptosteady energy separation

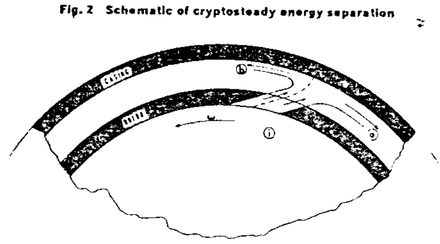


Fig. 3 External-separation arrangement

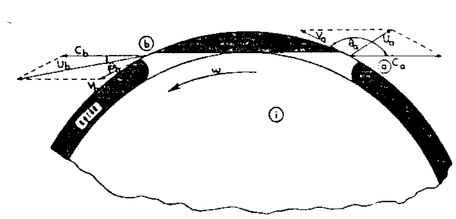


Fig. 4 Internal-separation arrangement

at the angular velocity which is required for the conservation of the total angular momentum of the flow in the laboratory frame of reference (frame F_a), thus producing the required motion of F_r relative to F_a.

Several variations of those arrangements are described in reference [12].

Cryptosteady energy separation was first proposed and analyzed in reference [11], which also contains an account of some of the experiments in which the validity of the concept was first tested and confirmed.

The analysis of reference [11] accounts for most of the pertinent parameters, including rotor torque and flow losses, but covers only situations in which the peripheral velocity, the discharge pressure, and the entropy rise are the same on the b as on the a side, the inclinations of the discharge velocities on the two sides are equal and opposite, and prerotation of the input flow is absent. The effects of departures from such symmetries have received relatively little attention until recently, except for a study by Hashem [13] on the effect of prerotation and for a series of performance analyses of internal-separation devices, in which the effects of prerotation (assumed to be uniform throughout the input flow) and of differences of nozzle inclination, peripheral velocity, and discharge pressure on the two sides have been individually examined by this writer.

A more comprehensive study of the subject has recently been completed by Graham [14], as part of a comparative analysis of the three classes of energy separation techniques—steady, nonsteady, and cryptosteady. In dealing with the latter technique, the Graham paper analyzes in detail the effect of unequal pressures on the behavior of the emerging jet in external-separation devices and also examines two output flow collection effects which are critical with these devices. Viscous reattachment of the deflected jets to the collector walls is found to be potentially beneficial, whereas flow pulsations—i.e., departures from cryptosteadiness—in the collection process, resulting from the use of confining vanes stationary in F_a, are found to be detrimental. The latter determination is of particular importance, in that it provides, for the first time, a firm rationale for focusing attention on those devices of this class in which the flow is truly cryptosteady.

For such devices, whether they be of the internal or of the external-separation variety, the Graham analysis develops "core performance" equations in which the most important design and operational parameters appear simultaneously, with full account of their nonlinear interactions. However, these equations require iterative solution in most cases, and their use is again limited in practice, because of their great complexity, to the individual evaluation of the separate effects of prerotation (again assumed to be uniform), rotor torque, and unequal back pressures, nozzle efficiencies, and exit flow orientations.

The present analysis approaches the same problem, for strictly cryptosteady situations, by a different route, which leads to simple, closed-form solutions in all cases. The analysis accounts for all design and operational parameters so far identified, as well as for their conceivable asymmetries (including unequal prerotations) and nonlinear interactions. Equations are also developed for the design of cryptosteady-flow energy separators in accordance with any given set of feasible performance specifications.

Generalized Performance Analysis

The following equations apply to both flows a and b:

$$h_e^* = h_i^0 - \frac{1}{2}\left(\frac{M}{r_e}\right)^2 + \frac{1}{2}\left(\frac{M}{r_e} - V_e\right)^2$$

$$= h_i^0 + \frac{1}{2}V_e^2 - M\omega$$

$$h_d^* = h_e^* + \frac{1}{2}(V_d^2 - V_e^2)$$

$$= h_i^0 + \frac{1}{2}V_d^2 - u'V_d \qquad (1)$$

and

$$c_d^2 = 2(h_d^* - h_d)$$

$$= 2\eta\left(h_d^* - h_i^0 + \frac{u_s^2}{2}\right) \qquad (2)$$

with

$$u_s^2 = 2h_i^0\left[1 - \left(\frac{p_d}{p_i^0}\right)^{\frac{\gamma-1}{\gamma}}\right] \qquad (3)$$

Equation (3) is plotted, for $\gamma = 1.40$, in Fig. 6.

From equations (1) and (2) there follows

$$c_d^2 = \eta(V^2 - 2u'V + u_s^2) \qquad (4)$$

equation (4) is plotted in Fig. 7.

Also,

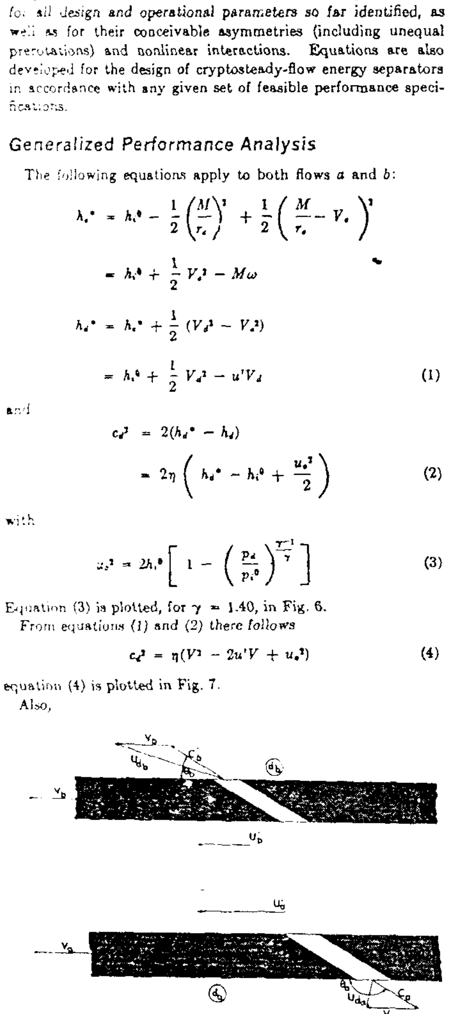


Fig. 5 Another internal-separation arrangement

$$u_d^2 = c_d^2 + V^2 + 2c_dV\delta \qquad (5)$$

and, by definition,

$$h_d^0 - h_d^* = \frac{1}{2}(u_d^2 - c_d^2) \qquad (6)$$

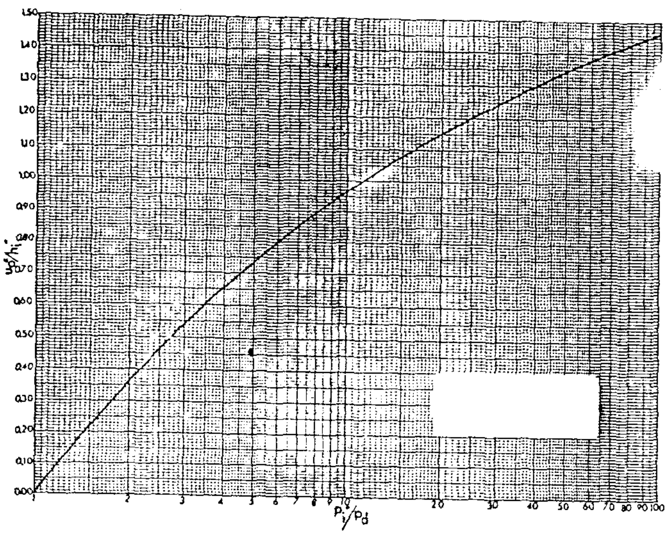
Equations (1), (4), (5), and (6) yield

$$\frac{h_d^0 - h_i^0}{V_d^2} = 1 + \delta\sqrt{\eta}\left[\left(\frac{u_s}{V_d}\right)^2 + 1 - 2\frac{u'}{V_d}\right]^{1/2} - \frac{u'}{V_d} \qquad (7)$$

The use of V as an independent variable is analytically convenient (as shown by the foregoing development) and is justified by the special constraints to which the selection of this parameter is subjected in practice (constraints of rotor size and structural strength, of bearing characteristics, etc.).

Equation (7) applies independently to each of the two outputs. It covers, therefore, such asymmetries as unequal peripheral velocities, discharge pressures, flow losses, prerotation velocities, and discharge angles. Rotor torque is implicitly accounted for through the mass flow ratio, as will be seen later.

Fig. 6



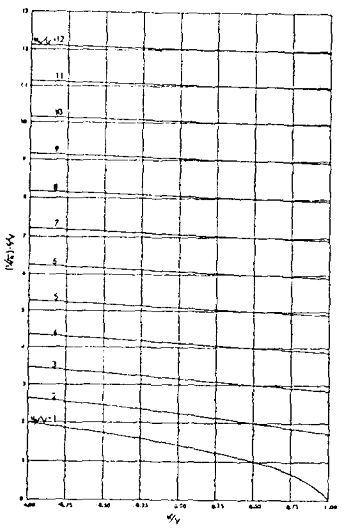


Fig. 7

The mass flow ratio is related to the specific enthalpy increments through the energy equation

$$\dot{m}_a h_a^0 + \dot{m}_b h_b = \dot{m}_i h_i^0 + L\omega$$

whence

$$\mu = \frac{h_i^0 - h_a^0 + \dfrac{L\omega}{\dot{m}_i}}{h_b^0 - h_i^0 - \dfrac{L\omega}{\dot{m}_i}} \qquad (8)$$

The "cold fraction" is

$$\nu = \frac{1}{1 + \mu} \qquad (9)$$

and the "cooling capacity coefficient" is, by definition,

$$\kappa = \nu(h_i^0 - h_a^0)/V_a^2 \qquad (10)$$

Equation (7) is plotted, for $\gamma = 1.40$, and for four different values of $|\delta\sqrt{\eta}|$, in Figs. 8(a) through 8(d). In each chart, the lower portion (for $\delta\sqrt{\eta} < 0$) provides the solution for the a side, and the upper portion for the b side. The notation u'/V on the abscissa scale stands for u'_a/V_{da} or u'_b/V_{db}, depending on whether this scale is used in conjunction with the a or b portion of the chart. If $|\delta_a\sqrt{\eta_a}| \neq \delta_b\sqrt{\eta_b}|$, the two stagnation enthalpy increments must be obtained separately, each from the appropriate chart.

Figs. 6 through 8 can be used, of course, also in the solution of the reverse problems resulting from interchanges of dependent and independent variables (e.g., in the determination of the rotor speed and input and discharge pressures that are required to produce a specified cooling capacity coefficient). Furthermore, visual inspection of these charts readily uncovers a good deal of useful information on the individual merits and relative importance of changes of various parameters in relation to their separate or combined effects on performance. Thus, for example,

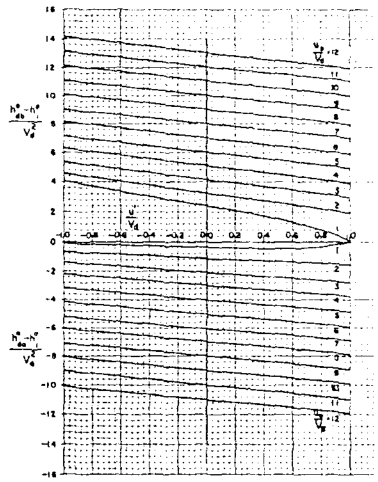


Fig. 8(a) Energy separator performance with $|\delta\sqrt{\eta}| = 1.0$

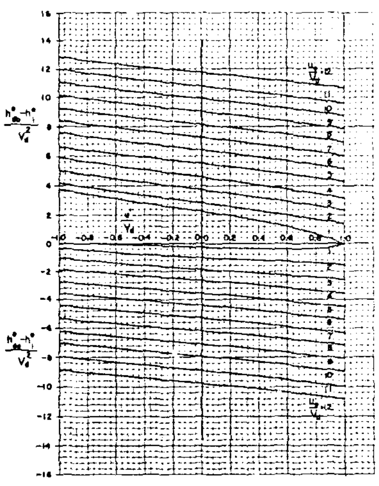


Fig. 8(b) Energy separator performance with $|\delta\sqrt{\eta}| = 0.9$

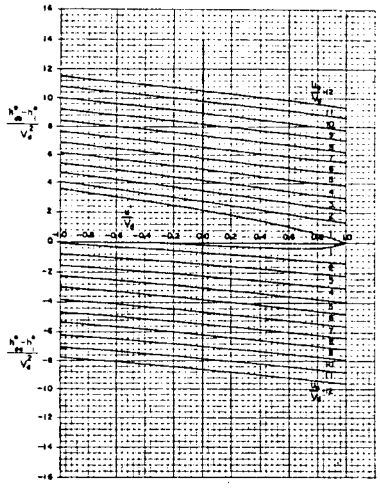


Fig. 8(c) Energy separator performance with $|\delta\sqrt{\eta}| = 0.8$

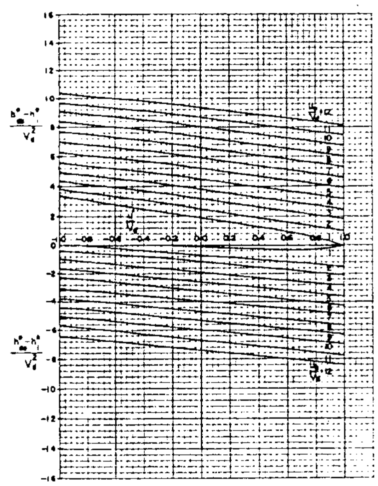


Fig. 8(d) Energy separator performance with $|\delta\sqrt{\eta}| = 0.7$

the charts confirm the existence of an optimum positive prerotation on the a side when the pressure ratio on that side is low; they reveal that opposite prerotations—positive on the a side and negative on the b side—can be remarkably beneficial from the standpoint of cooling capacity for any given rotor speed; and they provide a tool for the quick selection of the operational parameters that will best combine to produce any desired result.

The equations developed previously apply, of course, to both internal and external-separation devices. Thus, once a satisfactory solution has been identified, the determination of the combination of operational parameters (rotor speed, mass flow ratio, etc.) that will produce this solution is the same for both subgroups. The same cannot be said, however, of the manner in which the selected combination can be implemented. In the first place, in internal-separation devices the controlling design parameter is the nozzle area ratio α, whereas in external separation devices it is the impingement angle β. In the second place, the effect of unequal discharge pressures on rotor speed and mass flow ratio is markedly different in the two subgroups [14]. Finally, impingement wall boundary layer effects on performance, absent in internal separation, are believed to be potentially significant in external separation, although very little is yet known about them. The latter point is particularly important, in that it points to residual uncertainties that still make the correlation of design to performance a good deal less reliable with external than with internal separation. For this reason, only the internal-separation version will be considered in the following analysis of the controlling parameters.

Two cases will be discussed:

(a) the case in which the rotor nozzle flows are fully expanded on both the a and the b side, and

(b) the case in which the rotor nozzles are under-expanded and both flows are sonic at the nozzle exits.

Case (a). If the nozzle flows are fully expanded to prescribed pressures p_{da} and p_{db}, the velocities c_{da} and c_{db} can be obtained from equation (4) or from Fig. 7.

From the equation of state,

$$\frac{\rho_{db}}{\rho_{da}} = \frac{p_{db}}{p_{da}}\frac{h_{da}}{h_{db}}$$

$$= \frac{p_{db}}{p_{da}}\frac{h_{da}^* - \dfrac{c_{da}^2}{2}}{h_{db}^* - \dfrac{c_{db}^2}{2}} \tag{11}$$

and, from the definition of μ,

$$\mu = \alpha\frac{c_{db}}{c_{da}}\frac{\rho_{db}}{\rho_{da}} \tag{12}$$

Finally, equations (1), (11), and (12) yield

$$\alpha = \mu\frac{c_{da}}{c_{db}}\frac{p_{da}}{p_{db}}\frac{2h_i^0 + V_b^2 - 2u'_bV_b - c_{db}^2}{2h_i^0 + V_a^2 - 2u'_aV_a - c_{da}^2} \tag{13}$$

μ is calculated from equation (8), and it is through this parameter that rotor torque is accounted for.

Case (b). Since both flows are sonic, the mass flow ratio is [15]

$$\mu = \alpha\frac{p_{db}^*}{p_{da}^*}\left(\frac{h_{da}^*}{h_{db}^*}\right)^{1/2} \tag{14}$$

Now, on each side, $p_d^* = \lambda p_i^* = \lambda p_i^0\left(\dfrac{h_d^*}{h_i^0}\right)^{\frac{\gamma}{\gamma-1}}$. Thus

$$\mu = \alpha\frac{\lambda_b}{\lambda_a}\left(\frac{h_{db}^*}{h_{da}^*}\right)^{\frac{\gamma+1}{2(\gamma-1)}} \tag{15}$$

Equations (1) and (15) yield, for $\gamma = 1.40$,

$$\alpha = \mu\frac{\lambda_a}{\lambda_b}\left(\frac{2h_i^0 + V_a^2 - 2u'_aV_a}{2h_i^0 + V_b^2 - 2u'_bV_b}\right)^2 \tag{16}$$

where, as before, μ is obtained from equation (8) and accounts for L.

Symmetrical Cases

For those cases in which $\delta_b = -\delta_a$, $\eta_b = \eta_a$ (or $\lambda_b = \lambda_a$), $u_{ob} = u_{oa}$, $V_b = V_a = V$, $u_b' = u'_a = 0$, and $L = 0$, equations (8) through (10), and (13), or (16), yield

$$h_b^0 - h_i^0 = h_i^0 - h_a^0 + 2V^2 \tag{17}$$

and

$$(h_i^0 - h_a^0) = \mu(h_b^0 - h_i^0) \tag{18}$$

hence

$$h_i^0 - h_a^0 = \frac{2\mu}{1-\mu}V^2 \tag{19}$$

$$\mu = \alpha \tag{20}$$

and

$$\kappa = \frac{2\mu}{1-\mu^2} \tag{21}$$

Acknowledgment

This work was supported by the Mechanical Equipment Branch of the U. S. Naval Air Systems Command, under Contract Number N00019-72-C-0122.

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THERMODYNAMICS AND FLUID MECHANICS GROUP

AN INTRODUCTION TO THE DYNAMIC PRESSURE EXCHANGER

By P. H. Azoury, B.Sc., Ph.D.*

The historical background and operational principle of the Dynamic Pressure Exchanger (DPE) are outlined. The basic aerodynamic processes of cell-emptying and cell-filling are analysed by the 'method of characteristics' for air and for no temperature discontinuities in the unsteady flow pattern. The results of the analysis are then used to generalize performance qualitatively for overall pressure ratios up to the sonic threshold. It is shown that, for pressure wave effects to be fully utilized, a DPE rotor should run such that δ is of the order of or less than 0·5, where δ is the ratio of the time taken to open or close a cell to the time taken for a sound wave to travel a cell length at the thermodynamic stagnation state of the primary or secondary fluid. In the case where the thermodynamic properties of the fluids vary considerably, it is suggested that δ be referred to the gas which yields the highest sonic speed. In general, the extent to which the performance is affected by a change in δ, within the range $0 < \delta < 0{\cdot}5$, is inappreciable. It is also shown that the use of a transfer passage may be expected to yield a significant improvement in performance and an increased range in overall pressure ratio.

A number of applications are described and some recent developments are reviewed. It is also indicated that the main sources of loss can be incorporated within the method of characteristics used in the prediction of performance.

INTRODUCTION

PRESSURE EXCHANGE aims at providing means for the direct exchange of energy between flows which are initially at two different pressure levels; one fluid (the primary fluid) expands exerting its pressure forces to compress another fluid (the secondary fluid). The pressure limits of compression and expansion need not be equal. The fluids are in direct contact, there is no intermediary mechanical means such as pistons, compressors or turbines to accomplish the energy transfer.

Direct energy exchange can be effected by means of three known processes: (i) steady-flow transfer by mixing. To this mode, relevant to compressible or incompressible fluids, belongs the earliest type of pressure exchanger introduced by Knauff (1)† in 1906, called 'semi-static' pressure exchanger because its pressure characteristics are almost entirely independent of its speed of action; (ii) unsteady-flow transfer by means of pressure wave processes. This type of transfer, applied exclusively to compressible fluid flows, was first recognized in 1928 by Burghard (2). In this case, moving compression and rarefaction waves are utilized, effecting a more efficient transmission of energy than in (i) and considerably reducing mixing between primary and secondary fluids. Pressure exchangers that employ unsteady flow are known as 'dynamic' pressure exchangers; their pressure characteristics depend essentially on the rate of action of the device; (iii) 'crypto-steady-flow' transfer by means of moving pressure fields that are generated by the collision of two compressible or incompressible fluid flows. This type of transfer differs from type (ii) in that it does not require that the interacting flows be unsteady in all frames of reference. This method of energy exchange, first reported by Foa (3) (4) in 1955, is receiving increased attention (5).

This paper is relevant to only one type of pressure exchanger, the Dynamic Pressure Exchanger (DPE). However, the Semi-Static Pressure Exchanger (SSPE) also is described since it possesses some of the basic concepts of the DPE.

The MS. of this paper was first received at the Institution on 16th February 1965 and in its revised form, as accepted by the Council for publication, on 15th June 1965.

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† *References are given in Appendix II.*

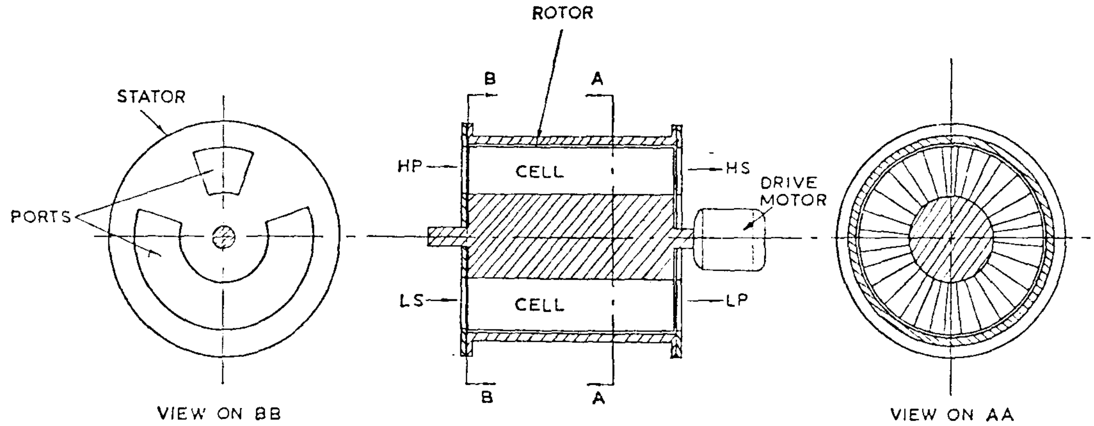


Fig. 1. Diagrammatic layout of pressure exchanger

In its usual form, employing type (i) or (ii) of energy exchange, the pressure exchanger consists of a cylindrical rotor with a plurality of straight axial cells arranged uniformly around its periphery, Fig. 1. The rotor rotates between two stators each of which has two ports to accommodate the two fluids.

Notation

- a Speed of sound.
- c 'No transfer passage' constant.
- c' 'One transfer passage' constant.
- c_p Specific heat at constant pressure.
- c_v Specific heat at constant volume.
- D Rotor diameter at mean radial cell height.
- d Cell width at mean radial cell height.
- h Radial cell height.
- k Constant.
- l Cell length.
- M Mach number in a cell end referred to isentropic static conditions in the port communicating with the cell.
- M^* Axial component of Mach number.
- $\dot{m}$ Mass flow rate.
- N Rotor speed, rev/min.
- p Stagnation pressure.
- p^* Static pressure.
- R Gas constant.
- T Absolute stagnation temperature.
- t Time.
- u Circumferential velocity of rotor at mean radial cell height.
- $\dot{V}$ Volumetric flow rate.
- x Space co-ordinate measured along flow.
- γ Ratio of specific heats c_p/c_v.
- δ Dimensionless cell width ad/lu.
- η_p Isentropic product efficiency = isentropic compression efficiency $\times$ isentropic expansion efficiency.
- μ Dimensionless net bulk input $(\dot{m}_{\mathrm{Hin}}T_{\mathrm{Hin}}-\dot{m}_{\mathrm{Hout}}T_{\mathrm{Hout}})/\dot{m}_{\mathrm{sw}}T_{\mathrm{Lin}}$.
- τ Dimensionless time at/l.
- ξ Dimensionless length x/l.

Subscripts

- 1 Condition in a cell after low-pressure scavenge.
- 1′ Condition in a cell before high-pressure scavenge for a DPE with one transfer passage.
- 2 Condition in a cell after high-pressure scavenge.
- 3 Condition in a transfer passage.
- 4 Condition in a cell before low-pressure scavenge for a DPE with one transfer passage.
- b Bleed.
- H High-pressure port or high-pressure region of cells.
- L Low-pressure port or low-pressure region of cells.
- M Medium pressure port.
- opt Optimum.
- sw Swept.

Figure notation

- H High-pressure fluid.
- HP High-pressure primary fluid.
- HS High-pressure secondary fluid.
- L Low-pressure fluid.
- LP Low-pressure primary fluid.
- LS Low-pressure secondary fluid.
- M Medium-pressure fluid.
- ⟶ Direction of fluid flow.
- ⟹ Direction of travel of cells.

SEMI-STATIC PRESSURE EXCHANGER

A diagrammatic representation of the SSPE for a refrigerating unit, first proposed in a 1928 patent by Lèbre (6), is used to indicate the operational principle of semi-static pressure exchange, Fig. 2.

The cells of the unit are shown folded out. Consider the cycle of events that occurs in any one cell. A cell leaving the low-pressure ports, which handle fluids LS and LP, contains secondary gas LS to be compressed. As the cell approaches the high-pressure port section, the cell gas undergoes a series of partial compressions by virtue of the expansion of the primary gas leaving that port section. A set of transfer passages is used to establish communication between primary and secondary fluid. High-pressure primary gas HP is now admitted into the cell and scavenges the secondary gas HS that has been compressed. The primary fluid now filling the cell undergoes a series of partial expansions (which serve to compress the secondary fluid) before it is discharged as cold gas LP, scavenged by LS. The cycle is repeated when the cell is completely filled with fluid LS. Make-up gas is used to compensate for the deficiency of volumetric flow rate of HS due to irreversibilities and to the temperature drop in the cooler. Fans are included in the flow circuit to aid in the scavenge operations.

The first commercial use of the SSPE became known in 1943 through the Swiss company of Brown Boveri (7). The application was that of a heat pump, for drying pur-

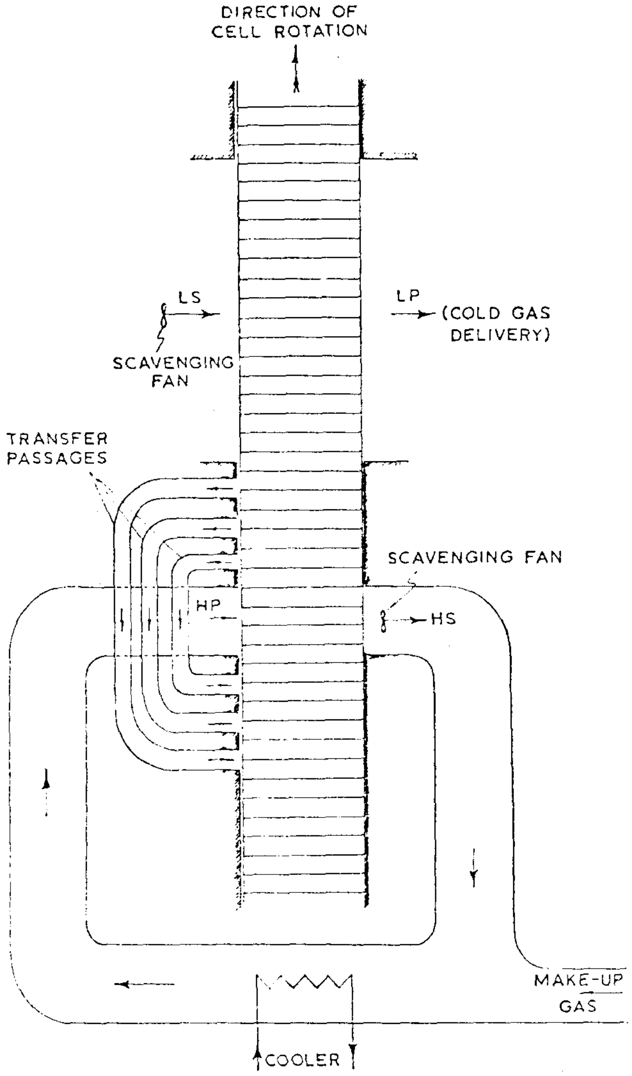


Fig. 2. Diagrammatic representation of Lèbre refrigerating SSPE showing cells folded out

poses in a paper mill, based on the patents of Lèbre (6) (8). In a more recent invention due to Poggi (9), an arrangement comprising two rotors in contra-rotation is suggested, the rotor cells being disposed radially as in the impeller of a centrifugal compressor. This feature is intended to facilitate the separation in the cells of the expanding primary fluid from the secondary fluid under compression, normally of different density.

DYNAMIC PRESSURE EXCHANGER

In a DPE, the basic ideas in the SSPE are still employed but the expansion and compression processes take place by means of pressure wave effects. These are of an unsteady flow nature.

Basic aerodynamic processes

In any DPE application, the aerodynamic processes in the rotor cells can be conceived as consisting, fundamentally, of cell-emptying and/or cell-filling.

The cell-emptying process

If a cell, containing a gas at a pressure higher than that of the surrounding atmosphere, is suddenly opened, a rarefaction wave is propagated into the cell, impelling gas out of the cell. In the simplified 'wave diagram' (i.e. plot representing the propagation of waves in space and time) of Fig. 3, the rarefaction wave is shown as a broken line; the actual 'fanning out' of the wave, from the moment the cell opens to the port, is shown in the wave diagram of Fig. 4*a*. In the cell-emptying process as best utilized in the DPE, the outlet port should be made as wide as possible, consistent with the avoidance of flow reversal in the port, and the maintenance of an adequately high outflow gas velocity during cell closing to prevent boundary-layer separation along the trailing edge of the port during the diffusion process. The residual cell stagnation pressure p_L is then at an optimum with respect to the port stagnation pressure p_M, with p_L always below p_M. The degree to which p_L falls below p_M is a measure of the cell-emptying performance.

Wave diagrams were drawn according to the 'method of characteristics' (10)–(14) for various values of initial pressure ratio p_H/p_M and 'dimensionless cell width' $\delta = 0, 0{\cdot}5, 1{\cdot}0, 1{\cdot}5$, for air ($\gamma = 1{\cdot}4$), zero cell wall thickness, no leakage, and no friction. In the computation of p_M, the diffusion efficiency in the outlet port was taken as 100 per cent. δ accounts for the cell-opening/closing time and is defined as the ratio of the time taken to open or close a cell to the time taken for a sound wave to travel a cell length at the thermodynamic stagnation state of the gas in the port. The port was made as wide as possible such that the average flow Mach number in it did not fall below the steady-flow Mach number when the cell was fully open. For the wave diagram shown in Fig. 4*a*, $\delta = 0{\cdot}5$, $p_H/p_M = 1{\cdot}79$ and $(p_L/p_M)_{opt} = 0{\cdot}50$.

Fig. 5 shows a plot of p_H/p_M against $(p_L/p_M)_{opt}$, with the 'sonic threshold' line indicated corresponding to $p_H/p_M = 1{\cdot}893$ for which the outflow velocity in the port is sonic when the cell is fully open. The plot shows that, over a wide range of initial pressure ratio, performance is more favourable with a finite than with an infinitely small cell-opening/closing time. This feature was first reported by Spalding (13). It is explained by the fact that, in the former case, advantage can be taken of the width of the pressure wave fan returning to the open end in cell-closing (Fig. 4*a*). It is found that the mean flow rate during the cell-opening/closing periods, when the cell is on the average only half-open, can exceed half that during the steady-flow condition when the cell is fully open. For an infinitely small cell-opening/closing time, however, the cell must be fully closed before the outflow velocity is reduced to that below the steady-flow value. A mean optimum value for δ lies at about 0·5.

If cell-emptying takes place with no diffusion, the static pressure at the open cell end, p^*_M, then equals the pressure of the surrounding atmosphere. In that case, the sole criterion of optimum port width is the avoidance of flow reversal. The resulting performance is represented by the curves of Fig. 6, the 'sonic threshold' line indicated corresponding to $p_H/p^*_M = 3{\cdot}583$.

In a SSPE, the cell-opening/closing operations take place relatively slowly, wave effects are slight, so that $p_L \simeq p_M$.

The cell-filling process

If a cell, containing a gas at a pressure lower than that of the surrounding atmosphere, is suddenly opened, a compression wave is propagated into the cell at supersonic speed, ahead of the interface separating port from cell gas. In the simplified wave diagram in Fig. 3, the cell-opening/closing time is infinitely small so that the compression wave is established instantly as a shock wave.

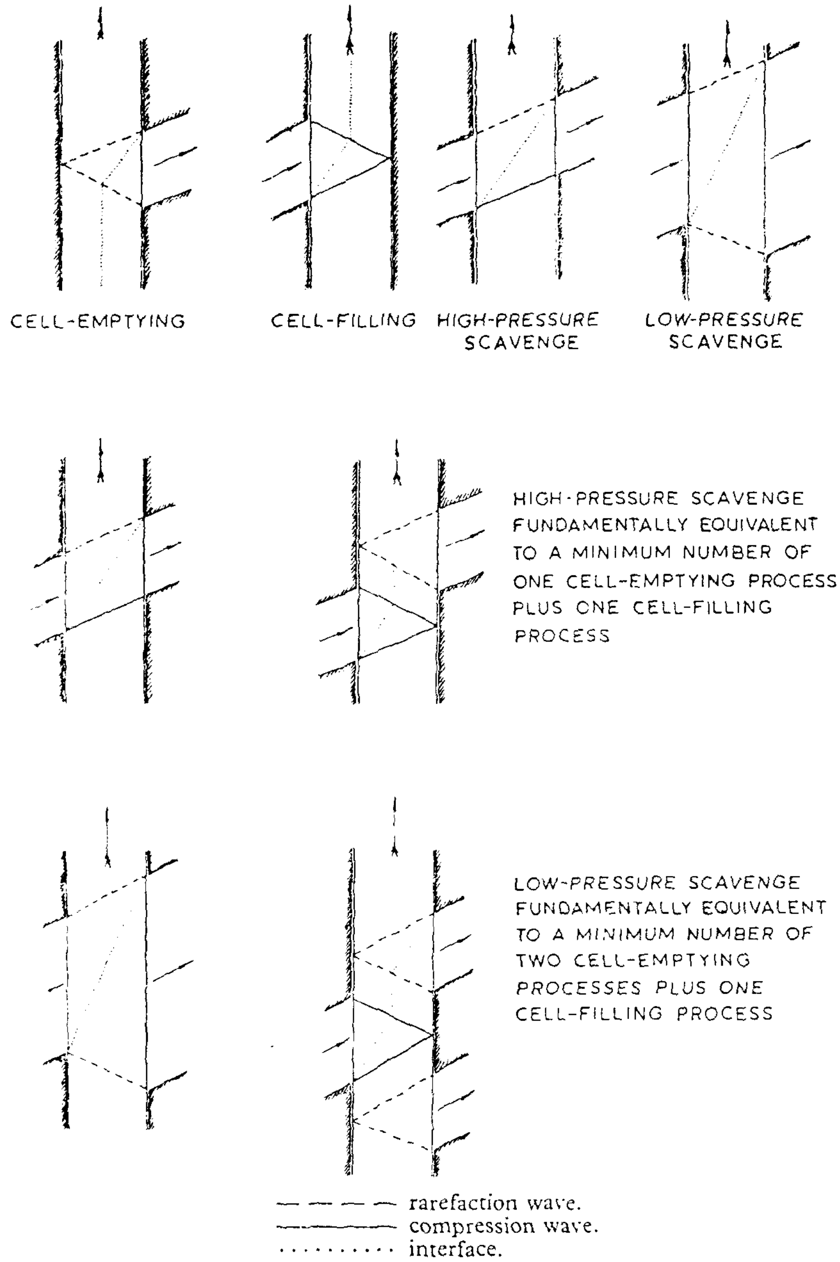
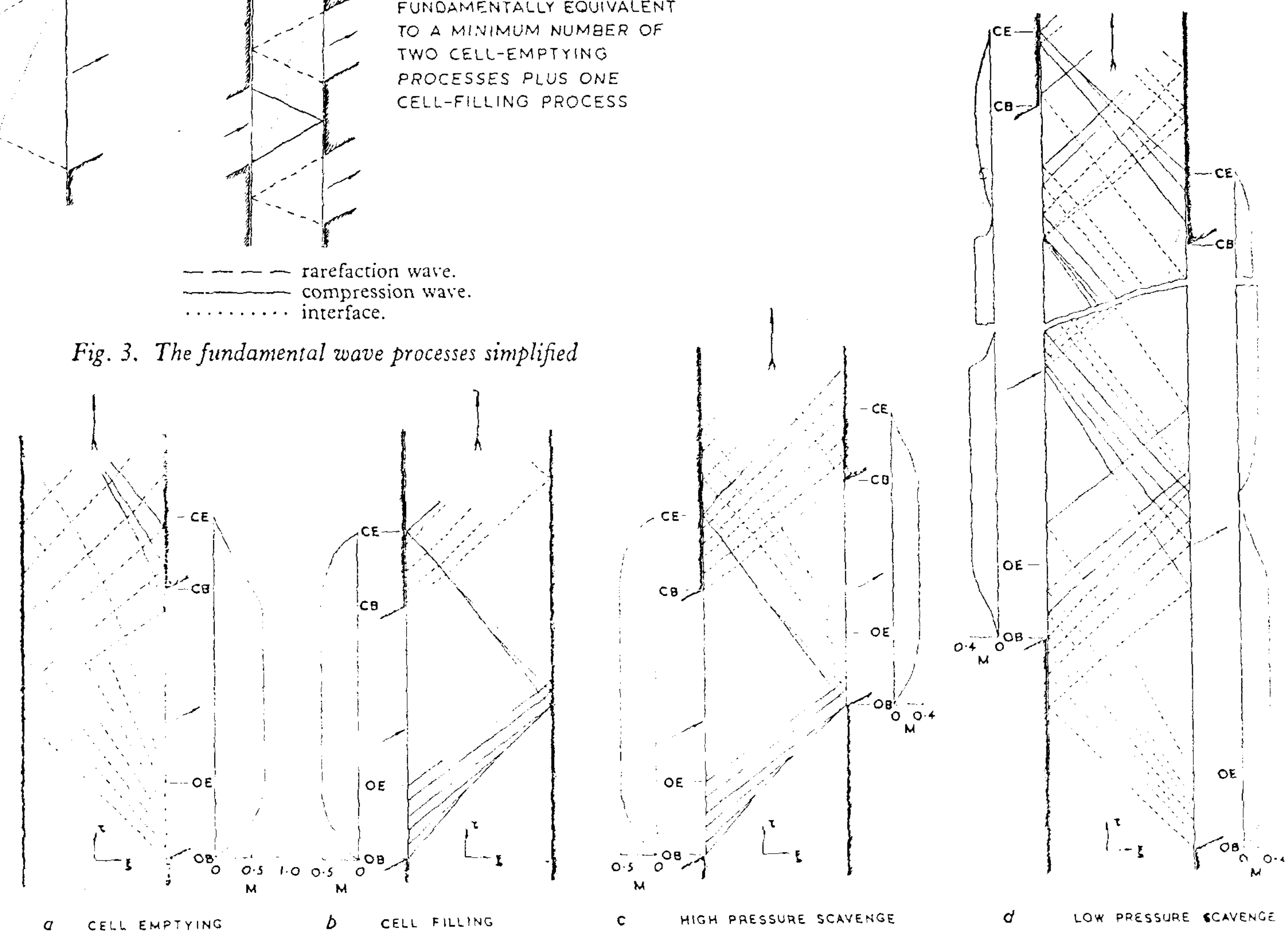


Fig. 3. The fundamental wave processes simplified

The actual coalescing of the compression wavelets into a shock wave, due to a finite cell-opening/closing time, is illustrated in Fig. 4*b*. In the cell-filling process as utilized in the DPE, the inlet port is normally made so wide that the cell is fully closed at the instant the shock wave, reflected from the closed end, reaches the open end. (A wider port would allow the shock wave to be reflected at the open end and initiate cell-emptying.) Under this condition, consistent with the avoidance of flow reversal, the maximum amount of port gas is trapped in the cell, thereby compressing the initial cell gas to an optimum pressure. The residual cell stagnation pressure p_H is then always higher than the port stagnation pressure p_M, the degree by which p_H exceeds p_M being a measure of the cell-filling performance. In a SSPE utilizing no pressure wave effects, both pressures are equal.

Wave diagrams are drawn for various values of p_M/p_L, $\delta = 0, 0{\cdot}5, 1$ and $1{\cdot}5$, for a cell initially filled with air ($\gamma = 1{\cdot}4$) that is isentropically expanded from the port stagnation state; under this condition and neglecting entropy gradients, the fluids on either side of the demarcation line separating port air from initial cell air, have equal temperatures, and waves encountering the demarcation



OB cell opening begins.
OE cell opening ends.
CB cell closing begins.
CE cell closing ends.

— — — — rarefaction line.
———— compression line.
.... interface.

Fig. 4. Wave diagrams of the basic DPE aerodynamic processes

line are wholly transmitted. Isentropic inflow, no leakage and no friction are also assumed. The port in each case was made as wide as possible, consistent with the avoidance of flow reversal. For the wave diagram shown in Fig. 4*b*, $\delta = 0{\cdot}5$, $p_M/p_L = 2{\cdot}45$ and $(p_H/p_M)_{opt} = 1{\cdot}60$. Fig. 7 shows a plot of p_M/p_L against $(p_H/p_M)_{opt}$ for the various values of δ, with the 'sonic threshold' line indicated corresponding to $p_M/p_L = 9{\cdot}017$ for which the inflow velocity through the fully open cell is sonic. It is seen that the effect of finite cell-opening/closing time is to deteriorate the cell-filling performance.

In a DPE, a cell-emptying process always follows, precedes or is combined with a cell-filling process for the pressure exchange operation to be cyclic. The depression in the cells after cell-emptying is subsequently used to induce port gas into the cells, i.e. to initiate cell-filling, and the super-port residual cell-pressure after cell-filling is then used to start a cell-emptying process. The DPE performance is better than that of the SSPE due to the superior residual pressure characteristics of the DPE cell-emptying and cell-filling operations.

The high-pressure scavenge process

In that part of a DPE where the secondary gas is to be compressed and scavenged by the primary gas, the cell-filling and cell-emptying processes may be combined into one process, the so-called 'high-pressure scavenge' process. The relevant diagram in Fig. 3 is shown for the condition of equal static pressures in the inlet and outlet ports and infinitely narrow cells; in that case, no rarefaction wave is generated and the shock wave is not reflected at the outlet port leading edge. In reality, with a finite cell width, the rarefaction wave is partly generated and the shock wave is partly reflected, even though the port static pressures may be equal. For full scavenge, the inlet and outlet port trailing edges should be so fixed that the demarcation line between the gases reaches the outlet port at the same time as the rarefaction wave generated by the closing of the inlet port. (The required number of cell-emptying and cell-filling processes to simulate full high-pressure scavenge increases with decreasing initial cell-filling pressure ratio and inlet/outlet port static pressure ratio.) For the wave diagram shown in Fig. 4*c*, δ (referred to the cell-filling port stagnation conditions) $= 0{\cdot}5$, the cell-filling initial pressure ratio $= 2{\cdot}45$, and the outlet port static pressure equals the inlet port stagnation pressure.

The low-pressure scavenge process

The primary gas, after compressing the secondary gas in the high-pressure scavenge section, is discharged, scavenged by the fresh, low-pressure secondary fluid that is induced into the cells as a result of the depression caused by the discharge. The entire process is known as the 'low-pressure scavenge' process and can be accomplished with a minimum number of one cell-filling and two cell-emptying processes, as indicated in Fig. 3. How-

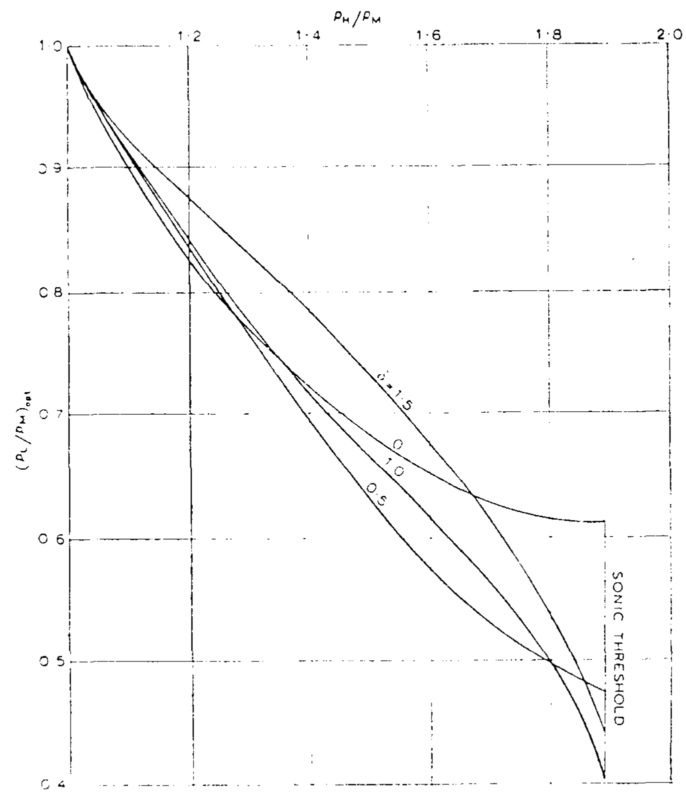


Fig. 5. Optimum cell-emptying performance, 100 per cent diffuser efficiency

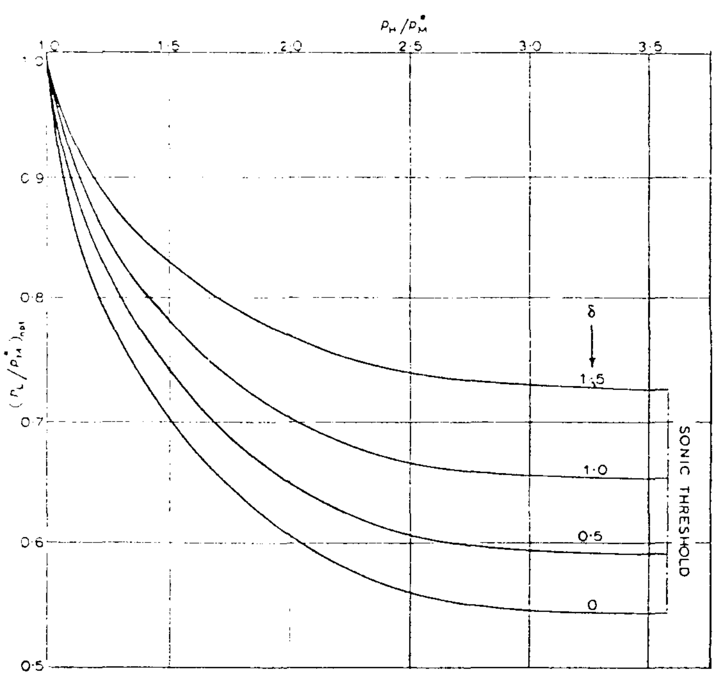


Fig. 6. Optimum cell-emptying performance, no diffuser

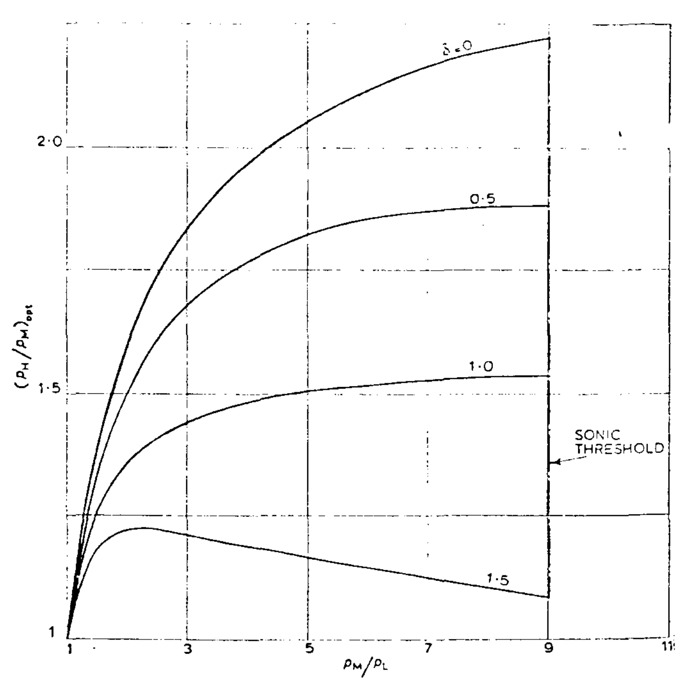


Fig. 7. Optimum cell-filling performance

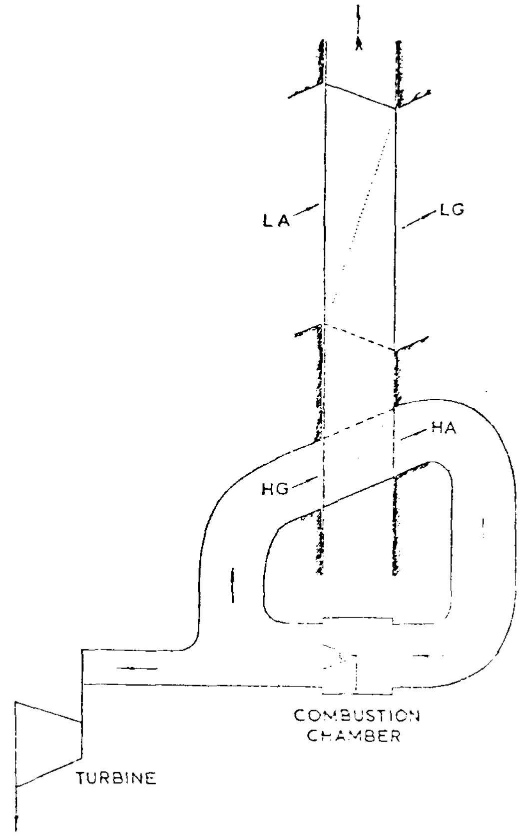
ever, advantage may be taken of the depression in the cells after the second cell-emptying process to induce additional fresh secondary fluid by means of a second cell-filling process and thereby pre-compress the charge prior to the main compression in the subsequent high-pressure scavenge section. The relevant cell-filling and cell-emptying processes need not take place separately; they may be combined such that only one inlet and one outlet port are required, as shown in Figs 3 and 4. The secondary gas inlet port opens at the instant the cell pressure at that end falls to the port pressure. For full scavenge with no pre-compression, the inlet port should close first such that the resulting rarefaction wave reaches the outlet end at the same time as the gas interface; the outlet port should then close at that time. For full scavenge with pre-compression, the outlet port should close first as soon as the gas interface reaches it; the inlet port should then close upon the arrival of the compression wave generated by the closing of the outlet port. (The required number of cell-emptying and cell-filling processes to simulate full low-pressure scavenge increases with decreasing initial cell-emptying pressure ratio and inlet/outlet port static pressure ratio.) The wave diagram shown in Fig. 4*d* was drawn for full scavenge with pre-compression, for δ (referred to the inlet port stagnation conditions) = 0·5, the first cell-emptying initial pressure ratio = 1·29, and for equal inlet port stagnation and outlet port static pressures.

OPERATIONAL PRINCIPLE OF THE DYNAMIC PRESSURE EXCHANGER

In general, the DPE may be used (i) in looped arrangements (with the high-pressure ports connected by ducting) acting, for example, as (*a*) a refrigerator or a heat pump, with the upper isobar being one of cooling and the high-pressure secondary fluid being energized by the addition of compressed make-up primary gas, as in the case of the Lèbre machine; (*b*) a gas generator, with the upper isobar being one of heating and some or all of the high-pressure secondary fluid being energized by means of a heat source, such as a combustion chamber of the gas-turbine type; (ii) in unlooped arrangements with no heat input or extraction. In case (ii) the DPE acts as a pressure interchanger. DPE units known as 'valved combustors' (15), in which periodic combustion takes place in the cells, are not considered in this paper.

The simplified operational principle of the DPE is indicated with reference to its use as a gas generator, Fig. 8. The cycle of operation is divided into the two basic scavenge processes previously described.

High-pressure hot gases HG coming from the combustion chamber are led to the high-pressure inlet port. The low-pressure cold air LA, initially in a cell before it comes into contact with that port, provides the necessary pressure differential to cause the flow. The air is compressed to approximately the same pressure level as that of the hot gases by a shock wave. The outlet port opens at the moment this wave reaches it. Beyond the high-pressure scavenge section, the cell contains only hot gases at a pressure level intermediate to that prevailing before compression and in the high-pressure ports. The cell is now opened on the right and the low-pressure scavenge process with pre-compression takes place. Upon its completion, only fresh air is trapped in the cell and the cycle is repeated



LA low-pressure fresh air.
LG low-pressure hot gases.
HA high-pressure fresh air.
HG high-pressure hot gases.

— — — — rarefaction wave.
———— compression wave.
· · · · · · · · · · interface.

Fig. 8. A simple DPE unit acting as a gas generator

when the cell is opened to the high-pressure hot gas port. Since the same cycle occurs in all the cells, the flow in the ports is substantially steady, even though the flow in the cells is unsteady.

The DPE unit shown diagrammatically in Fig. 8 can be used in two ways: (i) as a second stage for a gas-turbine unit, as proposed, for example, by Seippel (16), in which case the low-pressure inlet port would be connected to the compressor delivery and the low-pressure outlet port would be connected to the turbine inlet; (ii) as an engine 'in its own right', drawing its low-pressure air from the surroundings and supplying gas to a power turbine, for example. In both cases (i) and (ii), fluid for power production is bled from the hot gas flow in the DPE high-pressure loop (tempered, if necessary, with cold high-pressure secondary fluid), provided the hot gas temperature is above the idling temperature. For a given overall pressure ratio and peak temperature, the amount of this bleed-off is a measure of the pressure exchanger performance.

GENERAL DYNAMIC PRESSURE EXCHANGER PERFORMANCE

In reversible pressure exchange, no heat input or addition of make-up gas would be necessary to maintain a certain DPE overall pressure ratio, i.e. the compressed secondary gas leaving the high-pressure scavenge section would subsequently be used as high-pressure primary gas, and there would be no net 'bulk' input required in the unit. ('Bulk' is here defined as the product of mass flow rate and absolute temperature.) Due to irreversibilities, however, the bulk of secondary fluid discharged after compression ($\dot{m}_{H_{out}} T_{H_{out}}$) is always less than the bulk of primary fluid used to effect compression ($\dot{m}_{H_{in}} T_{H_{in}}$). Therefore, the net

bulk input $(\dot{m}_{H_{in}}T_{H_{in}}-\dot{m}_{H_{out}}T_{H_{out}})$ is a measure of DPE performance. This performance has been generalized qualitatively for air ($\gamma = 1{\cdot}4$), for overall pressure ratios up to the sonic threshold, and for no temperature discontinuities in the unsteady flow pattern, by considering the scavenge processes as combinations of cell-emptying and cell-filling, taken in the right order. The details of the steps involved are given in Appendix I for the two cases of a DPE with no transfer passage and with one transfer passage.

DPE performance with no transfer passage

The generalized performance curves for a DPE with no transfer passage are given in Fig. 9, where p_H/p_L is the overall pressure ratio, and μ is the dimensionless net bulk input defined as follows (the definition is due to Spalding (17)):

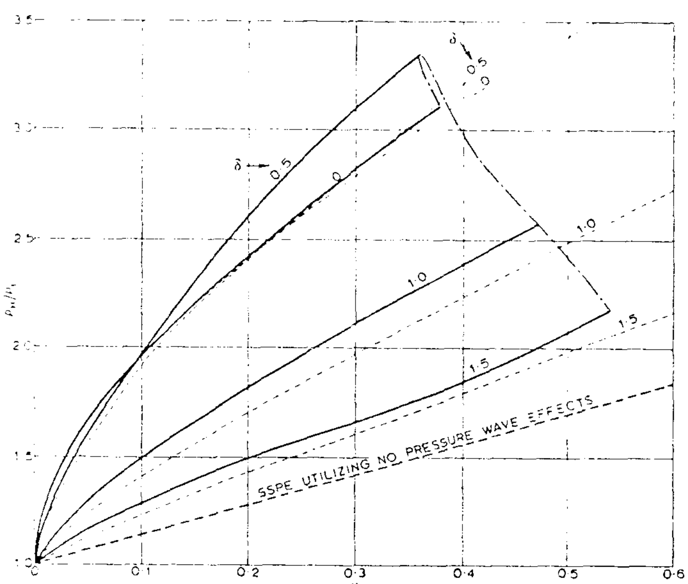
$$\mu = (\dot{m}_{H_{in}}T_{H_{in}}-\dot{m}_{H_{out}}T_{H_{out}})/\dot{m}_{sw}T_{L_{in}} \quad . \quad (1)$$

$\dot{m}_{sw}$ is the rotor swept mass flow rate referred to the inlet port stagnation conditions. Improvement in DPE performance is indicated if, for a given value of p_H/p_L, μ decreases, or if, for a given value of μ, p_H/p_L increases.

The simplified general performance equation is

$$p_H/p_L = 1+ck\gamma\mu \quad . \quad . \quad . \quad (2)$$

where c and k are constants (greater than unity) for any



- - - - - - - performance with low-pressure exhaust kinetic energy unavailable.

Fig. 9. Generalized performance for a DPE with no transfer passage

given value of δ. In the case of a SSPE utilizing no wave effects, the performance equation becomes

$$p_H/p_L = 1+\gamma\mu \quad . \quad . \quad . \quad . \quad (3)$$

The line representing equation (3) is included in Fig. 9 to indicate the superiority of the DPE over the SSPE.

DPE performance with one transfer passage

The generalized performance curves for a DPE with one transfer passage are given in Fig. 10. It is significant to note the increased range in p_H/p_L as a result of using a transfer passage.

The simplified general performance equation is

$$p_H/p_L = 1+c'k\gamma\mu \quad . \quad . \quad . \quad (4)$$

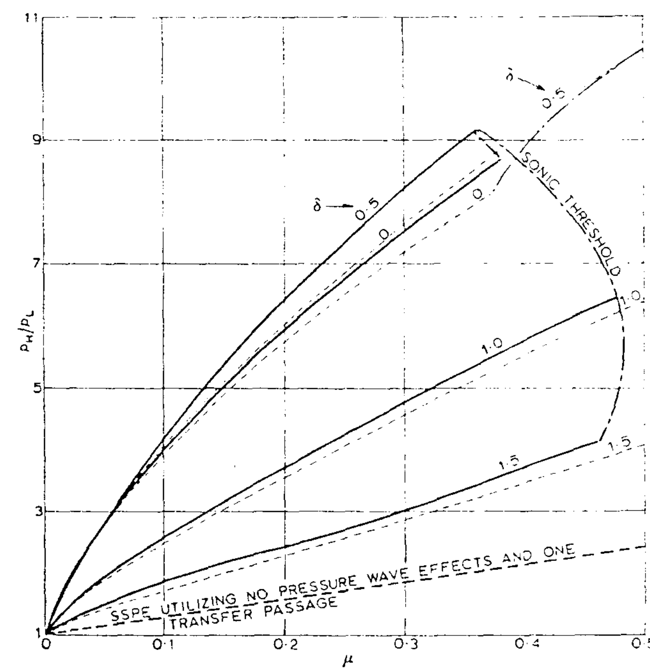
where k is the same constant as that in equation (2) and c' is estimated to be of the order of $2{\cdot}5c$ for all values of δ (see Appendix I). The effect of a transfer passage, therefore, is to multiply the 'no transfer passage' constant c by the estimated factor of 2·5. In theory, DPE performance improves with an increasing number of transfer passages. In practice, there is probably an optimum number, limited by the increased 'losses' associated with energy degradation in the passages, irreversible wave action, and leakage.

In the case of a SSPE with negligible wave effects, performance is given by

$$p_H/p_L = 1+2\gamma\mu \quad . \quad . \quad . \quad (5)$$

The line representing equation (5) is included for comparison in Fig. 10. It can be shown that, in general, the performance equation for a SSPE with n transfer passages is

$$p_H/p_L = 1+(n+1)\gamma\mu \quad . \quad . \quad . \quad (6)$$



- - - - - - - performance with low-pressure exhaust kinetic energy unavailable.

Fig. 10. Generalized performance for a DPE with one transfer passage

DPE performance with unavailable low-pressure exhaust kinetic energy

The generalized performance curves, shown in full lines, of Figs 9 and 10 are based partly on the assumption of 100 per cent diffusion efficiency in the high-pressure and low-pressure exhaust flows. In practice, with the outflow velocity decreasing across the low-pressure outlet port due to the drop in pressure ratio of the successive cell-emptying processes (see Fig. 4*d*), the diffusion process becomes inefficient and a large part of the exhaust kinetic energy is not recovered. The generalized DPE performance was re-estimated on the assumption that the low-pressure exhaust kinetic energy is unavailable, i.e. discharge is effected directly to the surrounding atmosphere. The steps involved are identical with those outlined in Appendix I, except that cell-emptying performance relevant to low-pressure scavenge was derived from Fig. 6 instead of Fig. 5. Figs 11 and 12 show the resulting generalized

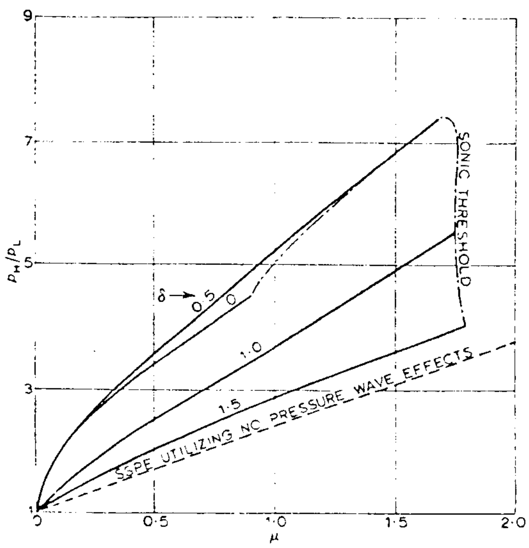


Fig. 11. Generalized performance for a DPE with no transfer passage and no low-pressure exhaust diffuser

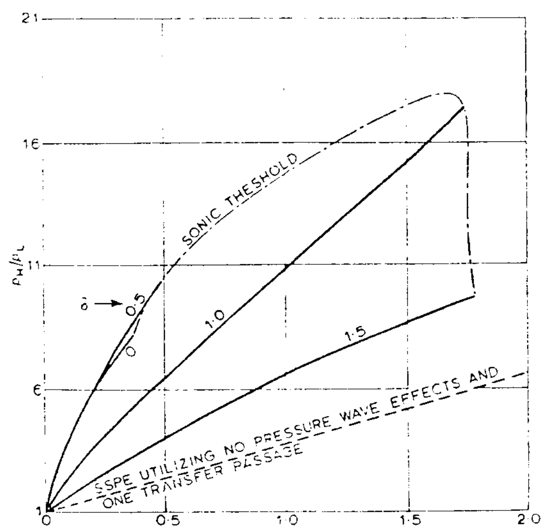


Fig. 12. Generalized performance for a DPE with one transfer passage and no low-pressure exhaust diffuser

DPE performance curves for no transfer passage and for one transfer passage, respectively. Parts of the curves are shown in Figs 9 and 10 to indicate the drop in overall performance. However, a comparison between the relevant curves shows that appreciably higher overall pressure ratios can be attained than are possible for the case of a DPE with full low-pressure exhaust kinetic energy recovery.

ROTOR SPEED AND DYNAMIC PRESSURE EXCHANGER PERFORMANCE

For a given rotor, utilizing gases of fixed thermodynamic properties, the cyclic wave pattern depends on the rotor speed, since the waves are generated as a cell end moves past a port edge. The rotor speed is thus directly related to the efficient operation of a DPE unit. It has been shown that in cell-emptying, an optimum value of $\delta \simeq 0{\cdot}5$ should be used, whereas in cell-filling, performance deteriorates with increasing δ. It is concluded, therefore, that for wave effects to be fully utilized, a DPE rotor should run such that δ is of the order of, or less than, 0·5. This result may be expected to hold also for the case where the thermodynamic properties of the primary and secondary flows vary considerably, since the qualitative effect of cell width on cell-emptying and cell-filling performance is independent of the gas or gases involved. In that case, however, it is suggested that the appropriate value of δ be referred to the gas which yields the highest sonic speed. Also, it is seen from Figs 9, 10, 11 and 12 that, in general, the extent to which the DPE performance is affected by a change in δ (which, for a given rotor, corresponds to a proportionate change in rotor speed) is unappreciable within the range $0 < \delta < 0{\cdot}5$.

In comparison with the conventional gas turbine, the DPE is a low-speed, and hence a more rugged, device. The resulting larger flow passages make the DPE less sensitive to gases containing solid suspensions.

SOME INDUSTRIAL APPLICATIONS OF THE DYNAMIC PRESSURE EXCHANGER

Gas generator

In a gas-generating DPE, the primary fluid is usually composed of hot combustion products and the secondary fluid of fresh air (Fig. 8). In order to overcome irreversibilities, such as those inherent in the wave processes and in the heating circuit, the gas-generating DPE, like the conventional gas turbine, requires a minimum heat input, the 'idling' heat input, to maintain a given overall pressure ratio. Unlike the gas turbine, however, the DPE idling heat input tends to zero as the pressure ratio tends to unity, since wave action is almost reversible at low pressure ratios. This signifies that the part-load performance of the DPE is basically superior to that of the gas turbine (18). Under idling conditions, the entire secondary fluid energized through combustion is re-circulated in the exchanger and no surplus fluid is thus available for power production. Additional heat transfer, corresponding to an increase in peak temperature, enables a portion of the hot

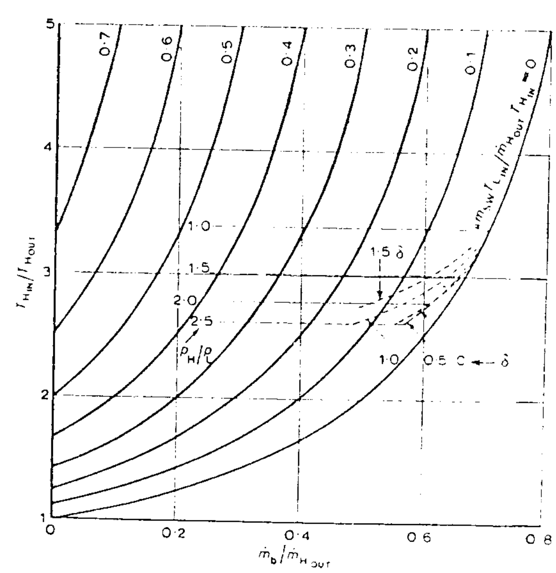


Fig. 13. DPE gas generator: peak temperature ratio versus bleed flow ratio with parameters of $\mu \dot m_{sw} T_{L_{in}}/\dot m_{H_{out}} T_{H_{in}}$

gases to be bled off and expanded through a turbine, say, to yield useful work. If the fuel mass flow rate is neglected, the bleed mass flow rate $\dot m_b$ becomes

$$\dot m_b = \dot m_{H_{out}} - \dot m_{H_{in}} \quad . \quad . \quad . \quad (7)$$

and the peak temperature ratio $T_{H_{in}}/T_{H_{out}}$ is obtained, by combining equations (1) and (7), as

$$\frac{T_{H_{in}}}{T_{H_{out}}} = \frac{1}{1 - \dfrac{\dot m_b}{\dot m_{H_{out}}} - \dfrac{\mu \dot m_{sw} T_{L_{in}}}{\dot m_{H_{out}} T_{H_{in}}}} \quad . \quad . \quad (8)$$

Fig. 13 shows a plot of the bleed mass flow rate ratio $\dot{m}_b/\dot{m}_{H_{out}}$ versus the peak temperature ratio $T_{H_{in}}/T_{H_{out}}$ for various values of the parameter $\mu\dot{m}_{sw}T_{L_{in}}/\dot{m}_{H_{out}}T_{H_{in}}$. $\dot{m}_{sw}/\dot{m}_{H_{out}}$ is a measure of the degree of scavenge; for full high-pressure and low-pressure scavenge, the ratio becomes unity. The optimum design pressure ratio is controlled by two relative effects: (i) at a constant design peak temperature $T_{H_{in}}$, $\dot{m}_b$ increases with decreasing p_H/p_L. This is illustrated in Fig. 13 by the δ curves (the gas generator performance following a constant δ line), based on the use of the full curves in Fig. 11 and equation (8), for $T_{H_{in}} = 1800°R$, $T_{L_{in}} = 530°R$, full scavenge, and isentropic compression of the low-pressure fresh air; (ii) the power developed per unit bleed mass flow rate increases with increasing p_H/p_L. These two effects have indicated that a simple DPE gas generator, i.e. unsupercharged or with no transfer passages, must operate at overall pressure ratios less than 2·5 for optimum performance (18). Low overall pressure ratios, common to all single-stage DPE's working under optimum conditions, signify low gas velocities which contribute to the erosion resistance potentialities of the device. Operation at overall pressure ratios above 2·5 is not known to lead to stall.

The first known working application of the DPE was as a gas generator (19) (20). This was probably in an attempt to exploit fully the ability of the DPE to permit high peak temperatures, higher than a conventional gas turbine can withstand. This is due to the fact that the rotor is alternately exposed to hot and cold gases. It is known that the rotor attains a temperature of the order of the mean between the hot gas and cold air inlets.

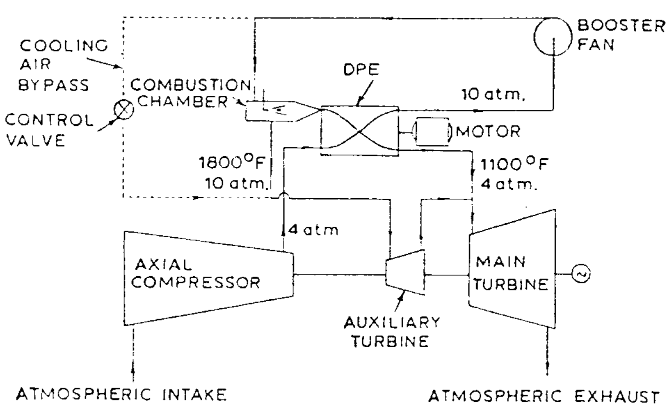


Fig. 14. Gas-turbine unit incorporating a DPE as proposed by Seippel

After the success of the Lèbre invention, the Brown Boveri Company followed up their interest in pressure exchangers by building, as a result of tests made on the first machine from 1941 to 1943, a DPE to serve as the high-pressure stage of a locomotive gas-turbine plant for British Railways, based on the patents of Seippel (16) (21). The relevant cycle diagram is shown in Fig. 14. So far as is known, the DPE unit worked satisfactorily but it was found that its use did not result in as high an improvement in performance (i.e. a power increase of 80 per cent together with an overall thermal efficiency increase of about 25 per cent) as had been anticipated. It was replaced by a heat exchanger which yielded a somewhat higher overall thermal efficiency. No further development work is known to have been made.

The inherent low-pressure ratio, and consequent low efficiency, of the simple gas-generating DPE may be increased by using a compound unit consisting of two exchangers, as suggested by Müller (22) for example, one supercharging the other, such that the supercharging unit operates preferably under idling conditions. This arrangement would then require only one power turbine. The cycle diagram of such a compound unit, incorporating a heat exchanger and fan, is shown in Fig. 15. The fan is

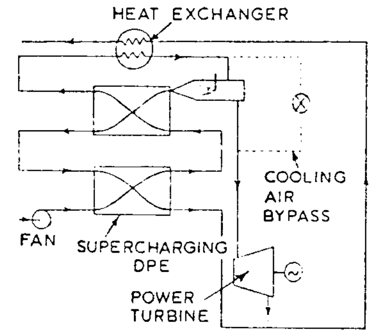


Fig. 15. A compound DPE unit acting as a gas generator

used for balancing the pressure loss in the heat exchanger. Preliminary investigations (18) have shown that this unit is highly competitive, both in design and in part-load performance. Moreover, there is hardly any increase in output and efficiency if the supercharging DPE also acts as a gas generator, requiring then an additional power turbine.

The overall pressure ratio may alternatively be boosted by means of transfer passages which are known to have been employed by Power Jets (Research and Development) Limited, other references to whose work appear below.

Supercharger

If the combustion chamber used in the gas-generating DPE is replaced by a reciprocating engine cylinder, the DPE would act, in conjunction with an appropriate control arrangement, as the engine supercharger. If the heat input required to maintain the desired supercharging pressure ratio is less than the available heat input, the DPE supercharger may also act as a gas generator, supplying high-pressure gases that would be put to some useful purpose, such as driving an ejector. Alternatively, the available supercharging pressure ratio may be reduced to the desired level by utilizing low-pressure scavenge with no pre-compression, resulting in a more compact unit.

Burghard (23) was probably the first to anticipate the supercharger application of the pressure exchanger as early as 1913. It was not, however, until recently that the principle was reduced to successful practice. In 1949 the ITE Circuit Breaker Company, Philadelphia, U.S.A., initiated a research and development programme to verify experimentally the calculated performance of dynamic pressure exchangers. An overall pressure ratio of 4·5 and an overall thermal efficiency of 16 per cent were achieved on a small test rotor, 4 in in diameter and 6 in long. Development was then focused on a DPE unit acting as a vehicle diesel-engine supercharger, the so-called 'Comprex'* (24) (25). The early version of the supercharger did not yield sufficient manifold pressure at very low engine speed, at which the clutch is engaged. The further development carried out at the Swiss Federal Institute of

* *'Comprex' is a trade name of the ITE Circuit Breaker Company.*

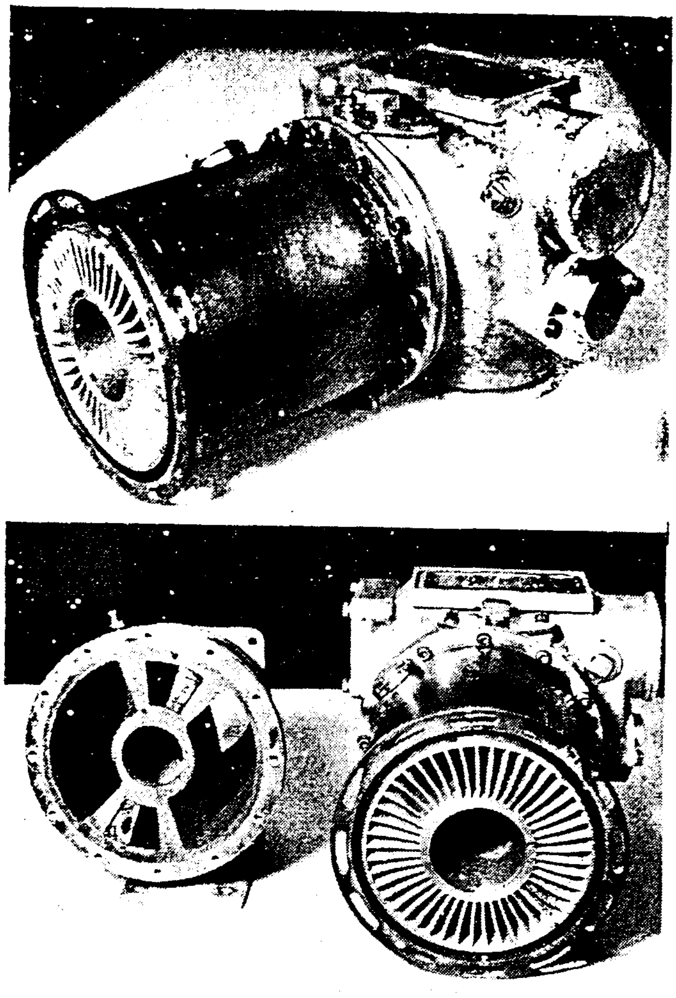
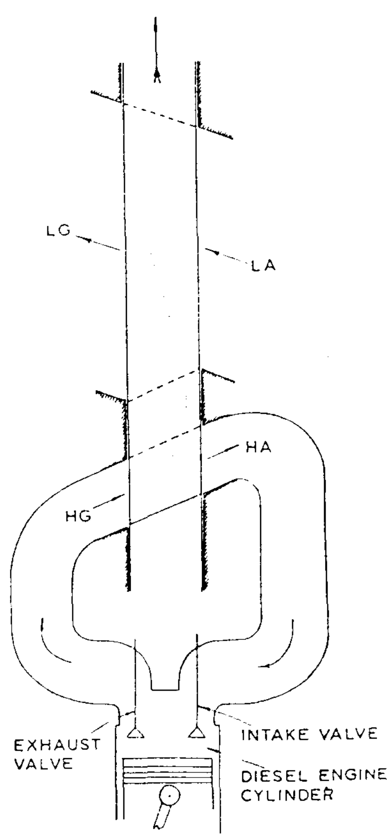


Fig. 16. The Comprex diesel supercharger
(Courtesy of the Swiss Federal Institute of Technology)

Technology, in collaboration with the Brown Boveri Company, led to cycle modifications overcoming this deficiency (**26**).

The Comprex supercharger is shown in Fig. 16; the unit operated on two cycle/rev. The relevant simplified wave diagram for each cycle is given in Fig. 17. The wave action is essentially the same as that of the Brown Boveri gas-generating DPE. It is important, however, to deliver pure air to the engine, and to ensure this, the high-pressure region is slightly 'underscavenged' (i.e. not all the compressed air is delivered to the engine) and the low-pressure region is slightly 'over-scavenged' (i.e. all the hot gases plus part of the induced fresh air charge is exhausted). Also, it is seen that the low-pressure port design is relevant to low-pressure scavenge with no pre-compression. This feature results in a reduced high-pressure port periphery, and consequently reduced port leakage, since the drop in the intake air pressure as a result of over-expansion yields an increased cell-filling pressure ratio in the high-pressure scavenge section, and hence an increased hot gas inlet velocity. In the case of the DPE gas-generating plant, these advantages would not overrule the drop in the general performance, as given by the p_H/p_L–μ relation, since low-pressure scavenge with no pre-compression depresses the factor c in the simplified overall performance equation (18) to a value below unity, worse than for the case of the SSPE.

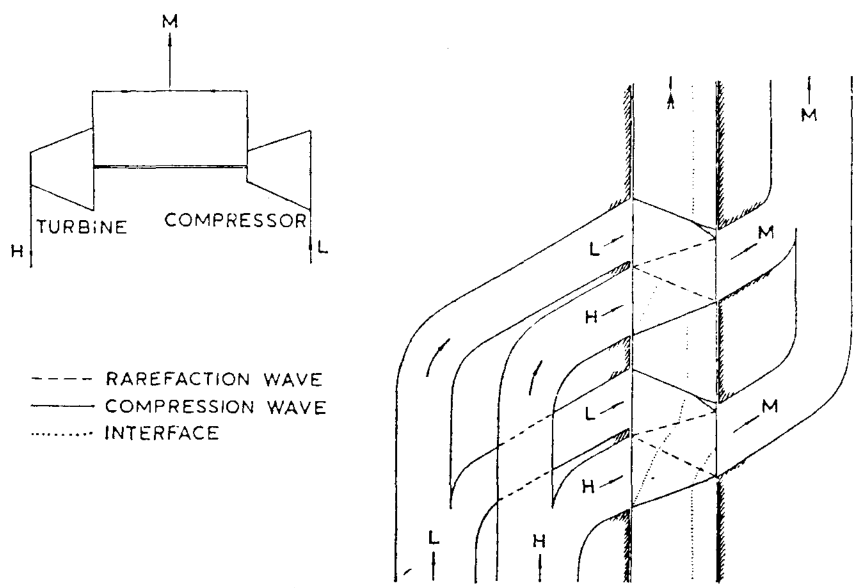
The superiority of the DPE supercharger over the conventional turbo-supercharger lies in its effectiveness being maintained down to very low speeds, its freedom from surge limits, and its immediate load response, since the increased exhaust energy is transferred instantly to the intake air. There is no lag due to inertia experienced with turbo-superchargers. Also, its efficiency is practically independent of scale, which makes it particularly advantageous in the small engine sizes (below about 100 hp) where turbo-superchargers suffer in efficiency. Among the advantages claimed by the ITE Circuit Breaker Company (**24**) (**25**), based on 20 000 miles of road testing with a diesel truck, are clean exhaust, fewer gearshifts, improved fuel consumption, no danger of over-speed and low sensitivity to unbalance.



LA low-pressure fresh air.
LG low-pressure hot gases.
HA high-pressure fresh air.
HG high-pressure hot gases.

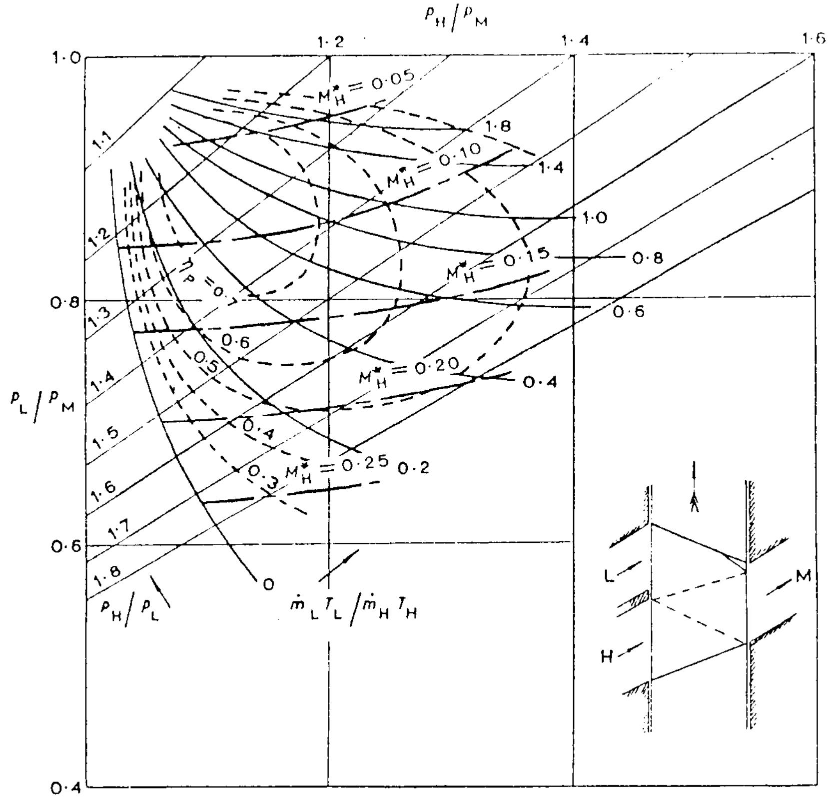
– – – – rarefaction wave.
——— compression wave.
· · · · · · interface.

Fig. 17. DPE acting as a diesel engine supercharger



a equalizer, utilizing compressor–turbine set. *b* equalizer, utilizing a two cycle DPE.

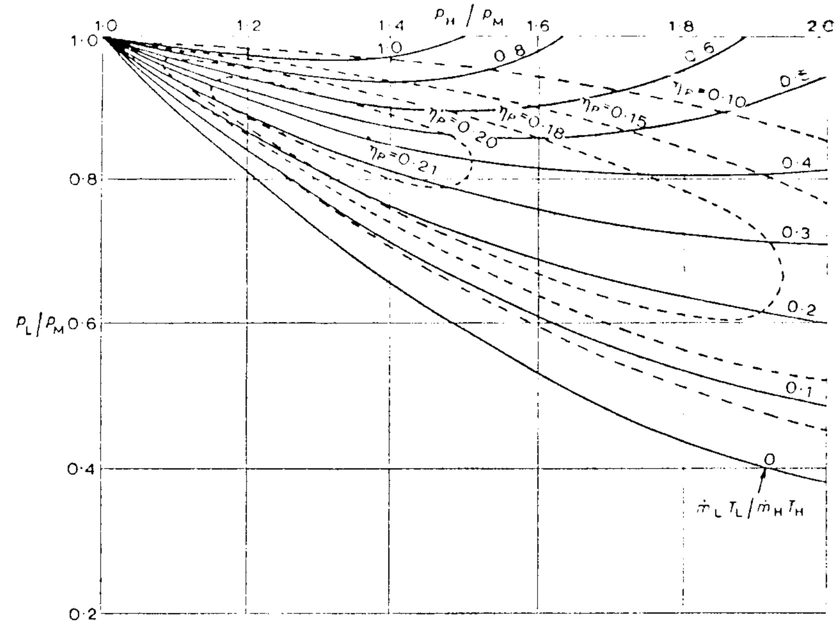
Fig. 18. Equalizer



Experimental conditions:

$T_H = 308°K$, $p_L \simeq 1$ atm., $T_L/T_H \simeq 0{\cdot}95$
Rotor speed = 5500 rev/min
H port width = 47°
M port width = 48°
L port width = 43°
phasing = 24°

Fig. 19. Experimentally obtained equalizer performance with air as working fluid



'g. 20. Experimentally obtained ejector performance with air at room temperature as working fluid

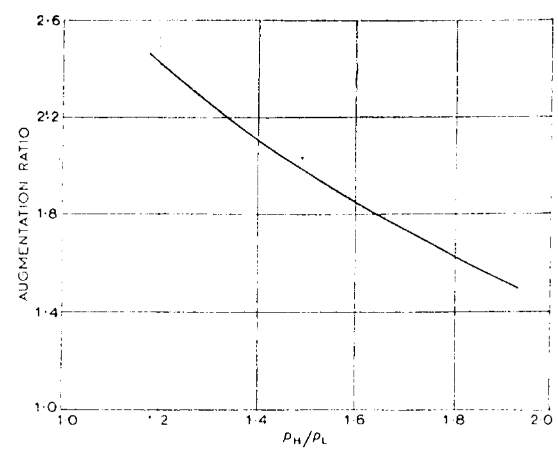
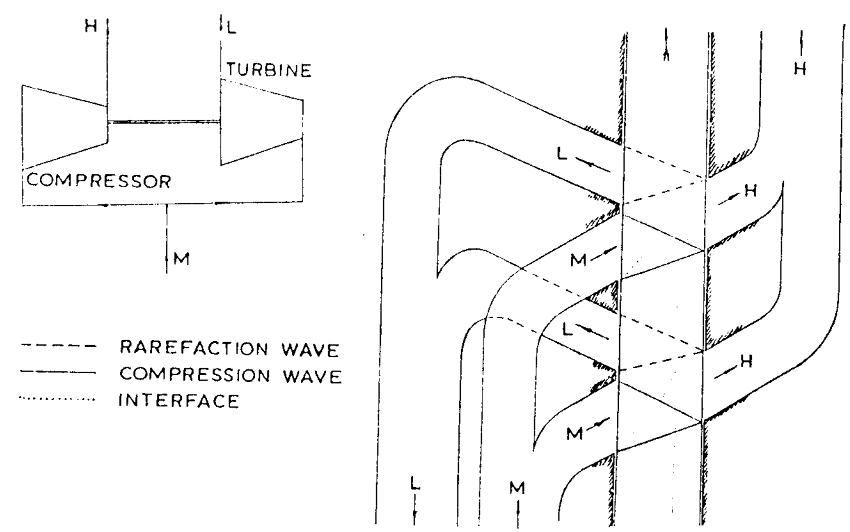


Fig. 21. Estimated performance of equalizer working as a thrust augmenter using efflux of a jet engine as a source of driving fluid, $T_L = 520°R$, $T_H = 1935°R$



a divider, utilizing compressor–turbine set.

b divider, utilizing a two cycle DPE.

Fig. 22. Divider

Air compressor

The gas-generating DPE unit can be used alternatively as a source of compressed air supply by bleeding fresh air instead of hot gases from the upper isobar. Some degree of under-scavenge in the high-pressure section and over-scavenge in the low-pressure section, as in the supercharging unit, would be essential to maintain a delivery of pure compressed air.

Refrigerator

The DPE operating as a refrigerator is basically the same as the Lèbre unit (6). Because of wave processes, however, the DPE unit with no transfer passages would have as high a coefficient of performance as the Lèbre semi-static unit with several transfer passages. There has been no published data concerning the performance of a DPE refrigerator. It is known, however, that Power Jets (Research and Development) Ltd, who have been interested in pressure exchangers since 1949, are now in an advanced stage of development of a DPE unit for a variety of industrial applications. The superior performance of the DPE refrigerator over that of non-condensable gas refrigeration plants has been established on the Power Jets test unit. The most promising application of the DPE refrigerator is as a mine cooler and a Power Jets prototype has already been supplied for service in a deep gold mine.

Equalizer/divider

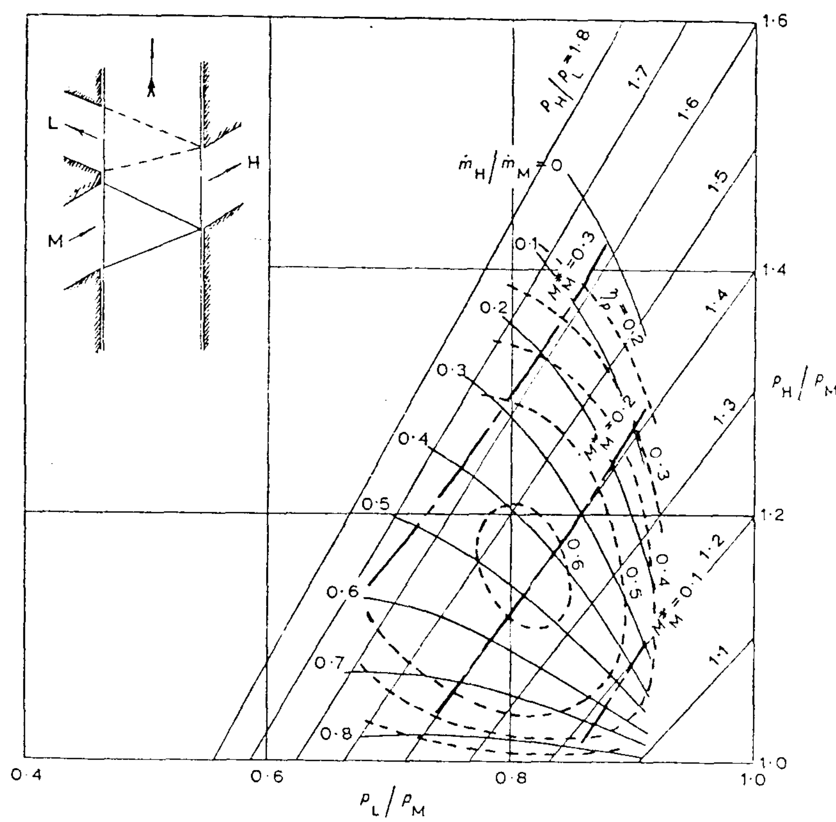
The equalizer/divider is one example in which the DPE acts purely as a pressure interchanger. In the so-called equalizer (27)–(31), a high-pressure gas stream is expanded to the same pressure level as that to which a low-pressure stream of the same or a different gas is compressed, the two exhaust streams being then led to a common duct. The circuit diagram using a conventional compressor–turbine set is shown in Fig. 18*a* and the simplified wave diagram in the case of a DPE is shown in Fig. 18*b*. Fig. 19 shows the experimental performance curves obtained by Kentfield (31) on a Power Jets test unit with air as working fluid. At suitable pressure ratios, the results indicate a performance that is superior to that of a conventional steady-flow ejector (32) (Fig. 20). The maximum isentropic product efficiency η_p (product of isentropic compression and expansion efficiencies) for the ejector is 21 per cent and for the equalizer about 75 per cent. The plot shown in Fig. 21 represents an estimate of the equalizer thrust augmentation potential, using the efflux of a jet engine as a source of driving fluid, with $T_H = 1935°R$ and $T_L = 520°R$. It is assumed that the performance characteristics are identical with those displayed in Fig. 19, the effect of the hot driving fluid being taken into account by the parameter $\dot{m}_L T_L/\dot{m}_H T_H$.

The so-called divider (28)–(31) is equivalent to the equalizer working in reverse, i.e. a single entering stream of medium pressure is divided into a high-pressure stream and a low-pressure stream. If a compressor–turbine set is used, the circuit diagram would be as shown in Fig. 22*a*; Fig. 22*b* shows the simplified wave diagram if wave processes are utilized in a rotor. The divider may be used as a pressure booster in cases where quantities of a gas at a pressure higher than that of the main supply are desired. Fig. 23 shows the performance curves experimentally obtained by Kentfield (31) on the Power Jets test unit. A photographic view of the machine is given in Fig. 24. The pressure-boosting application would be uniquely suitable in cases where erosion problems may hinder the use of turbo-machines. The cooling potential of the divider low-pressure stream is illustrated by the experimental performance curves of Fig. 25 due to Kentfield (31). An application in which good use is made of the DPE erosion resistance potentialities is that of the divider working as an expansion engine in a vapour-compression refrigerator (Fig. 26).

THE DYNAMIC PRESSURE EXCHANGER AS A TOOL FOR HIGH-SPEED AERODYNAMIC RESEARCH

In the laboratory, the DPE possesses the usefulness of the shock tube as a tool for high-speed aerodynamic research (a DPE cell approaching the high-pressure scavenge section is operated like the driven tube of a shock tube) with the advantage of a more sustained period of operation. This feature of the DPE has recently been exploited at Cornell Aeronautical Laboratory, Inc., Buffalo, New York, in the so-called 'wave superheater'. The prototype unit (33) produces a hypersonic flow of superheated gas for a duration of as long as 30 seconds, greatly exceeding the useful duration of up to 15 msec yielded by a shock tunnel (34).

The wave superheater is basically a DPE acting as a pressure interchanger with an overall pressure ratio that may exceed 100, so that the primary fluid (hydrogen or



Experimental conditions:

$T_M = 308°K$, p_L = 1 atm.
Rotor speed = 6000 rev/min
H port width = 48°
M port width = 45°
L port width = 45°
phasing = 24°

Fig. 23. Experimentally obtained divider pressure-boosting performance with air as working fluid

helium) can compress and heat the secondary fluid (air and/or argon) to extremely high temperatures. The secondary fluid then expands through a hypersonic nozzle to a test section. A coolant gas is used to maintain the rotor temperature to an acceptable level. Also, a 'priming' gas, introduced into the cells prior to the induction of the secondary fluid, establishes a uniform flow through the cells and thereby ensures a cyclic action of the device (Fig. 27). The full-scale unit (35) (36) (37), now in operation, can deliver uncontaminated air at stagnation conditions up to 7000°R and 110 atmospheres. Air flows up to 8 lb/s, covering the range of Mach 2–15, can be attained for test periods up to 15 s. It is estimated that if argon is substituted for air as a test gas, the unit can superheat 12 lb/s of argon at 17 000°R. The wave superheater rotor is shown in Fig. 28.

The wave superheater is believed to be the only facility in existence capable of realistically simulating the extreme conditions of hypersonic flight at low altitudes (< 150 000 ft). Approximately full-scale models of some hypersonic vehicle nose cones can be tested in the wave superheater tunnels under the aerophysical conditions associated with launching, gliding and re-entering.

SOME SOURCES OF LOSSES IN DYNAMIC PRESSURE EXCHANGERS

Assuming that the port widths and phasings are 'tuned' to the relevant wave pattern, the major sources of 'losses' in the DPE are due to the following effects.

Irreversible wave action

Shock waves are non-isentropic, although the entropy increase becomes significant only for pressure ratios across the shock higher than about 2. In general, the industrial applications of the simple DPE must operate at single-stage overall pressure ratios less than 2·5 for optimum performance, so that entropy increments in that case may be neglected. Rarefaction waves are isentropic but they cause wave interactions, which lead to irreversibilities, through their tendency to fan out (see Fig. 4). The wave pattern is made more complex when the gases on either side of an interface have different temperatures, molecular weights, or ratio of specific heats. Waves encountering such a discontinuity are transmitted as well as reflected in such a manner that the pressure and flow velocity on either side of the interface are equal. These effects can be treated

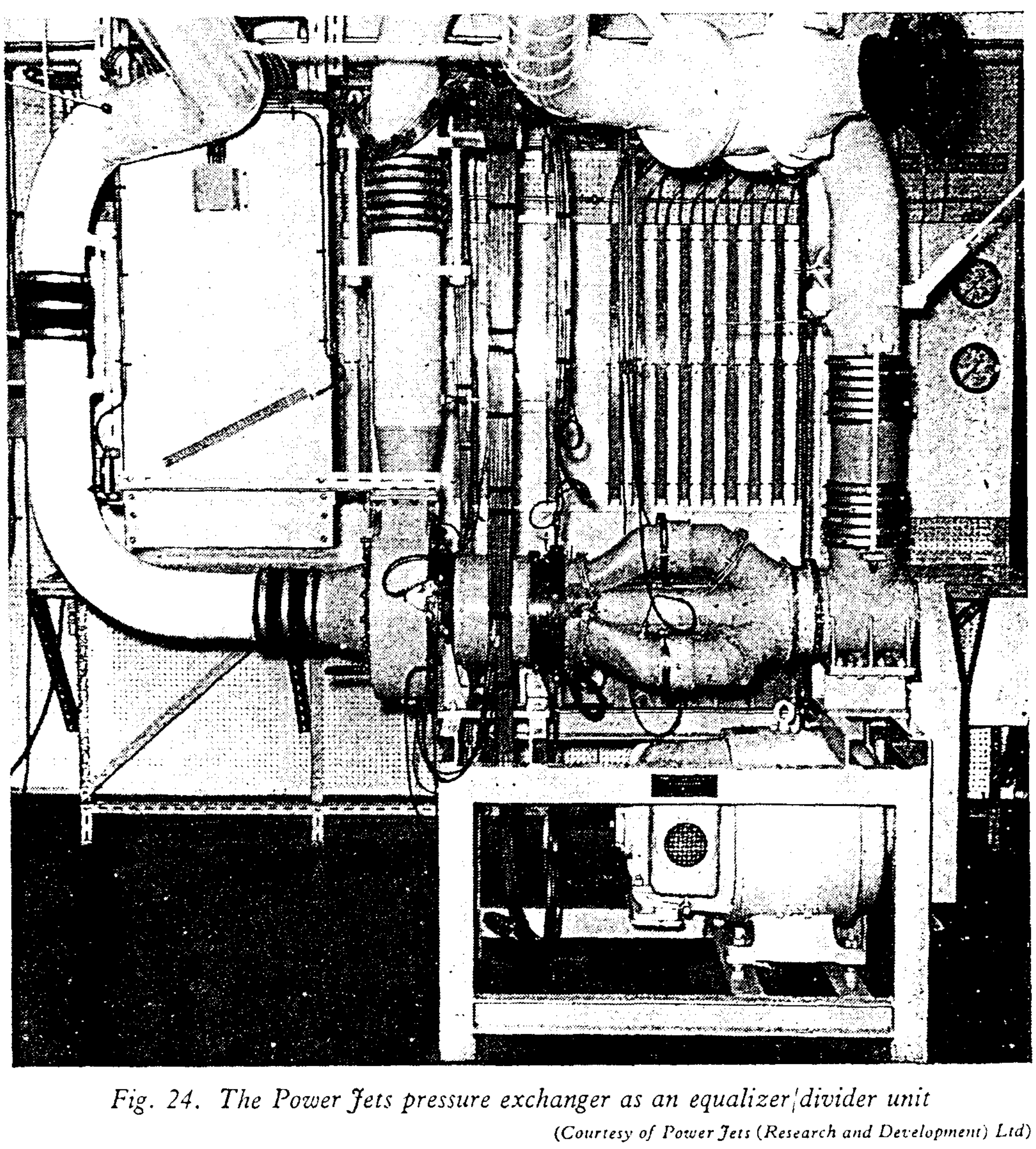


Fig. 24. The Power Jets pressure exchanger as an equalizer/divider unit

(Courtesy of Power Jets (Research and Development) Ltd)

theoretically through simple modifications in the method of characteristics (11) (38) (39).

Finite cell width

The finite cell-opening/closing time (δ in dimensionless form) gives rise to irreversibilities through causing the pressure waves to be more spread out. Finite cell width is usually accounted for in wave diagrams by means of Jenny's approach (10) based on the assumption that the flow is one-dimensional in the main body of the cell, while at the opening or closing end the flow has the steady-state configuration corresponding to the instantaneous geometry. Refinements in this method may be made by accounting for cell-wall thickness (31) and by applying appropriate values of discharge coefficients. It has been noted that the general DPE performance is inappreciably affected by finite cell-opening/closing time provided that δ is equal to or less than about 0·5; in the case where the thermodynamic properties of the primary and secondary flows vary considerably, it is suggested that δ be referred to the gas which yields the highest sonic speed. The maximum values of δ given in Table 1, together with the relevant rotor dimensions and speeds, refer to a number of DPE units mentioned above.

Table 1. Details of different DPE units

Type	D (in)	d (in)	h (in)	l (in)	N (rev/min)	Maximum δ
Full-scale wave superheater .	60	0·55	1·43	66	2700	0·08
Prototype wave superheater .	12	0·25	0·25	16	4500	0·29
Power Jets equalizer . .	6·48	0·59	2·15	11	5500	0·39
Comprex super-charger c 110 .	3·23	0·19	1·10	4·3	13 400	0·43

Leakage

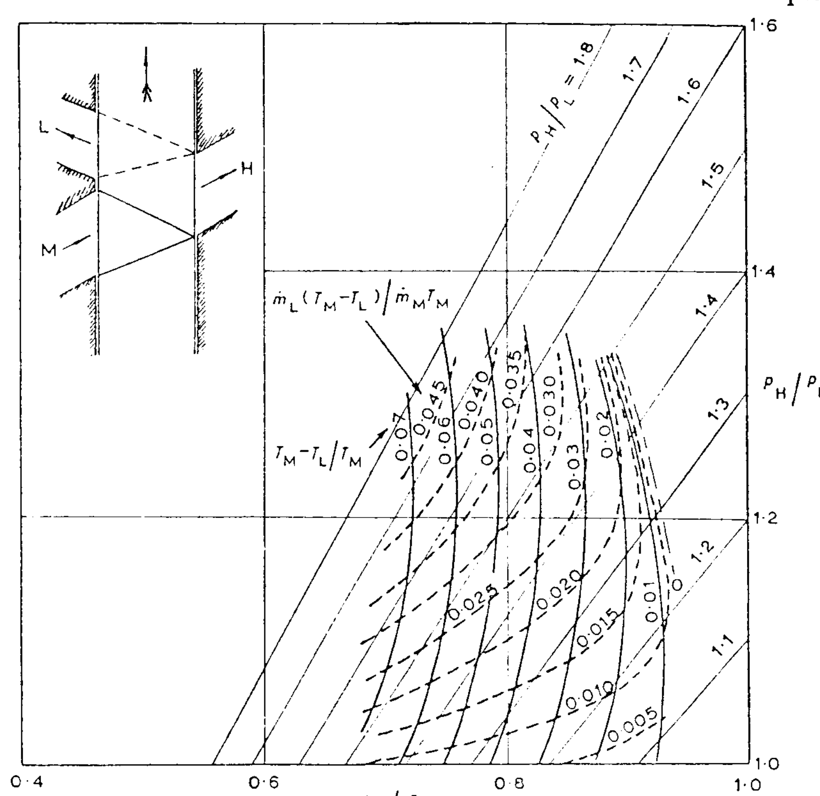
The rotor–stator clearance should be kept small, under static as well as running conditions, otherwise the loss in the strength of the waves and circumferential as well as radial leakage, at the high-pressure ports in particular, would have a serious effect on the DPE performance. In the Power Jets equalizer/divider, for example, a clearance of the order of 0·007 in is maintained.

In the theoretical treatment of leakage effects, a 'closed' cell end may be considered as an isentropic nozzle whose throat area is equivalent to the effective clearance gap area (40). In predicting the effects of leakage on the equalizer/divider performance, Kentfield (31) presents a general theory, to account for circumferential and radial leakage in the vicinity of ports, based on an approach by Spalding (41) and on a paper by Kearton and Keh (42).

Friction

For a given cell height and length, friction effects increase with decreasing cell width, but on the other hand wave action tends to be less irreversible, since compression and rarefaction waves are generated more quickly. Thus, for a given rotor, running at a certain speed and utilizing gases of fixed properties, there is an optimum cell width, and consequently an optimum number of cells, in the light of the relative effects of friction and pressure waves.

The mechanism of friction in unsteady gas flow is highly complex due to the fact that the flow history should be



Experimental conditions:
$T_M = 308°K$, $p_L \simeq 1$ atm.
Rotor speed = 5000 rev/min
H port width = 48°
M port width = 45°
L port width = 45°
phasing = 24°

Fig. 25. Experimentally obtained divider cooling performance with air as working fluid

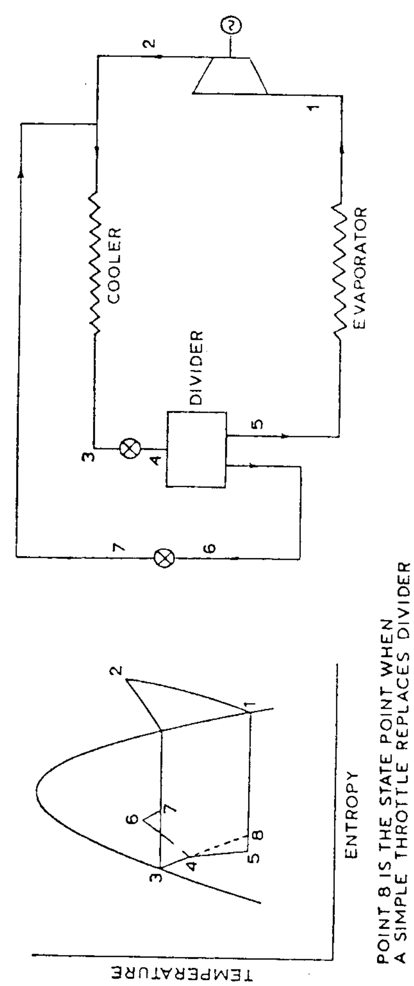


Fig. 26. Vapour-compression refrigerator utilizing a DPE divider

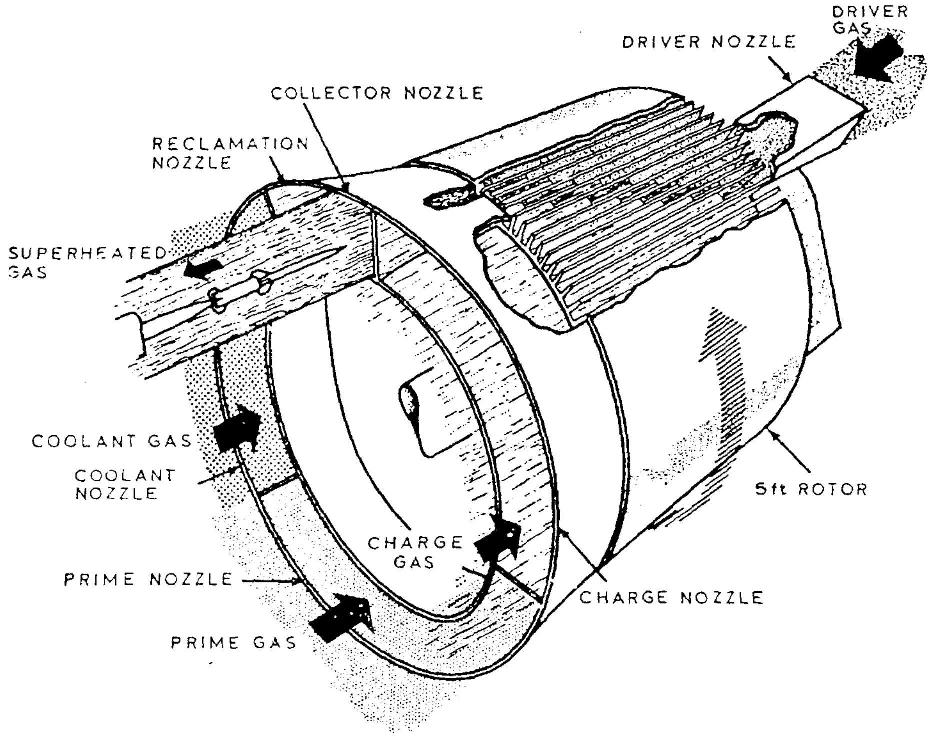


Fig. 27. Wave superheater rotor and nozzles

(*Courtesy of Cornell Aeronautical Laboratory, Inc.*)

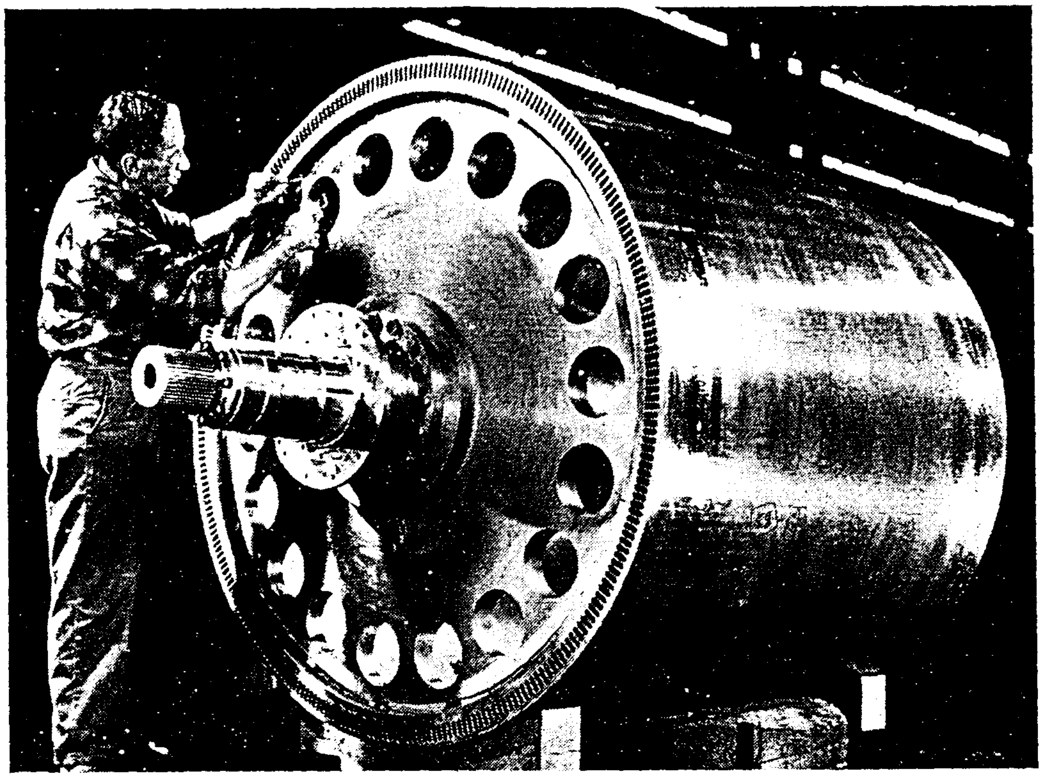


Fig. 28. Rotor for wave superheater hypersonic tunnel

(*Courtesy of Cornell Aeronautical Laboratory, Inc.*)

taken into account. Several investigators (43) (44) (45) have indicated rigorous procedures for the inclusion of wall friction in a wave diagram. The calculations involved are laborious, unless they are handled by a digital computer (46). The comparative insignificance of friction effects in the DPE has been indicated in the case of the Comprex supercharger (40) and the Power Jets equalizer/divider unit (31).

Mixing, diffusion and heat transfer

The undesirable form of energy transfer through mixing and diffusion always takes place across the interface of two different fluids in a DPE cell. Mixing takes place also due to the flow pattern in the opening and closing processes of a cell end and, for gases of different densities in a cell, due to centrifugal acceleration which causes interpenetration of the gases (1 [illegible] This latter effect is similar to that of 'gravity waves' between two incompressible fluids in an open channel. Heat transfer is another source of entropy increment and takes place across the interface as well as between the gases and the rotor. It is due to heat transfer, on the other hand, that the rotor temperature remains substantially below the peak gas temperature.

Mixing, diffusion and heat transfer apparently have no serious effect on DPE performance. In a Power Jets test unit, for example, an equivalent compression efficiency of 85 per cent, for an overall pressure ratio of 2, is reported (47). Satisfactory agreement (with a maximum deviation of 10 per cent) between experimental and theoretical results, based on the method of characteristics, was reached (48) by neglecting the effects of mixing, diffusion and heat transfer, as well as those due to friction and entropy increase across shock waves, for overall pressure ratios below 2·5.

Kinetic energy losses

Significant kinetic energy losses may be present in two locations, due to inefficient diffusion of the discharged gas flow: (i) in the high-pressure outlet port: good diffuser design in conjunction with the proper choice of port edge angles would render this loss tolerable; (ii) in the low-pressure outlet port. For the order of the single-stage overall pressure ratios ($< 2{\cdot}5$) utilized in the industrial type of DPE, the low-pressure scavenge ports are several times (e.g. five times) as wide as the high-pressure scavenge ports, for the same degree of scavenge. This results in inefficient diffusion of the low-pressure exhaust flow, even with the proper choice of port-edge angles and diffuser configuration. Some of the exhaust gas kinetic energy may be recovered by slightly bending the blades, thus making the rotor partially or totally self-driving. This was first suggested by Knauff (49) in 1906. A more recent patent relevant to the DPE is by Berchtold (50). However, the low-pressure exhaust-gas kinetic energy may be required to overcome the pressure drop in a cleaner and silencer incorporated in the low-pressure circuit.

Rotor drive

The power required to drive the DPE rotor for overcoming bearing friction and windage losses is normally small. In the Comprex supercharger, for example, the required drive power is about one per cent of the engine output (25).

CONCLUSIONS

The DPE is an entirely novel type of machine which effects direct energy exchange between compressible fluid flows by means of unsteady-flow transfer, utilizing pressure wave processes. Energy exchange by mixing is present only to a minor extent. The wave processes are related directly to the rate of action of the device. This rate, therefore, is a governing factor of DPE performance. Analysis of the basic wave processes by the 'method of characteristics' for air ($\gamma = 1{\cdot}4$) and for no temperature discontinuities in the unsteady flow pattern, yielded the following results:

(1) For pressure wave effects to be fully utilized, a DPE rotor should run such that δ is of the order of or less than 0·5. In the case where the thermodynamic properties of the primary and secondary flows vary considerably, it is suggested that δ be referred to the gas which yields the highest sonic speed.

(2) In general, the extent to which the DPE performance is affected by a change in δ, within the range $0 < \delta < 0{\cdot}5$, is inappreciable.

(3) The use of a transfer passage yields a significant improvement in DPE performance and an increased range in overall pressure ratio.

Since it contains the mechanisms of a compression-expansion engine, the DPE may be utilized to perform the same functions as those of a conventional compressor-turbine set. Moreover, the following potential advantages are obtained:

(1) Ability to permit higher peak temperatures due to the fact that, when exposed alternatively to hot and cold gas flows, the rotor attains an intermediate temperature.

(2) Superior part-load performance since wave action tends to be reversible with decreasing pressure ratio.

(3) Better erosion resistance due to the relatively large flow passages and low gas velocities.

(4) No stall characteristics.

In the laboratory, the DPE possesses the usefulness of the shock tube as a tool for high-speed aerodynamic research with the advantage of a more sustained period of operation.

DPE performance, flow conditions, optimum port widths and phasings can be predicted with a satisfactory degree of precision by means of the method of characteristics.

ACKNOWLEDGEMENTS

The author wishes to thank Mr T. G. Hicks, Managing Director of Power Jets (Research and Development) Ltd, for permission to publish information relevant to the Power Jets pressure exchangers, Professor D. B. Spalding, with whom he had many stimulating discussions on the subject, Dr C. Seippel for permission to include the background of the Brown Boveri gas-generating unit, and Professor M. Berchtold for the information regarding the development of the Comprex diesel supercharger.

The author wishes also to express his indebtedness to the Swiss Federal Institute of Technology for making available Fig. 16, Cornell Aeronautical Laboratory, Inc., for Figs 27 and 28, and Dr J. A. C. Kentfield for Figs 20 and 21 and for permission to publish Figs 19, 23, 25 and 26 abstracted from his Ph.D. thesis.

APPENDIX I

GENERALIZED PERFORMANCE FOR A DPE WITH NO TRANSFER PASSAGE

Application of the First Law of Thermodynamics to control surface c (Fig. 29*a*), assuming no leakage, no heat transfer, and constant and common specific heats for both primary and secondary gases, yields

$$c_p(\dot{m}_{H_{in}}T_{H_{in}}-\dot{m}_{H_{out}}T_{H_{out}}) = c_v(\dot{m}_2T_2-\dot{m}_1T_1) \quad . \quad (9)$$

or

$$\gamma(\dot{m}_{H_{in}}T_{H_{in}}-\dot{m}_{H_{out}}T_{H_{out}}) = \dot{m}_2T_2-\dot{m}_1T_1 \quad . \quad (10)$$

where c_p = specific heat at constant pressure, c_v = specific heat at constant volume, $\gamma = c_p/c_v$, $\dot{m}$ = mass flow rate, T = absolute stagnation temperature.

In order to generalize it, the net bulk input $(\dot{m}_{H_{in}}T_{H_{in}}-\dot{m}_{H_{out}}T_{H_{out}})$ is made dimensionless and is denoted by μ which is defined as follows:

$$\mu = (\dot{m}_{H_{in}}T_{H_{in}}-\dot{m}_{H_{out}}T_{H_{out}})/\dot{m}_{sw}T_{L_{in}} \quad . \quad . \quad (11)$$

where $\dot{m}_{sw}$ = the rotor swept mass flow rate referred to the inlet port stagnation conditions, i.e.

$$\dot{m}_{sw} = p_{L_{in}}\dot{V}_{sw}/RT_{L_{in}} \quad . \quad . \quad . \quad (12)$$

where p = stagnation pressure, $\dot{V}_{sw}$ = the volumetric flow rate swept out by the cells, R = the gas constant.

From equations (10) and (11), we have

$$\gamma\mu\dot{m}_{sw}T_{L_{in}} = \dot{m}_2T_2-\dot{m}_1T_1 \quad . \quad . \quad . \quad (13)$$

Since $\dot{m}_2T_2 = p_2\dot{V}_{sw}/R$ and $\dot{m}_1T_1 = p_1\dot{V}_{sw}/R$, equation (13) may be re-written as

$$\gamma\mu = (p_2-p_1)/p_{L_{in}} \quad . \quad . \quad . \quad . \quad (14)$$

which is one form of the general DPE performance equation.

It is more convenient, however, to correlate μ to the DPE overall pressure ratio $p_{H_{out}}/p_{L_{in}}$, since the cell stagnation pressures p_1 and p_2 are difficult to measure. It is first supposed that both high-pressure and low-pressure scavenge performances may be predicted from the cell-emptying and the cell-filling performance curves of Figs 5 and 7. This is done in a step-by-step manner in which a scavenge operation is considered as a combination of cell-filling and cell-emptying, taken in the right order. Full high-pressure scavenge is assumed to be accomplished with one cell-emptying and one cell-filling process, and full low-pressure scavenge is assumed to be accomplished with two cell-emptying

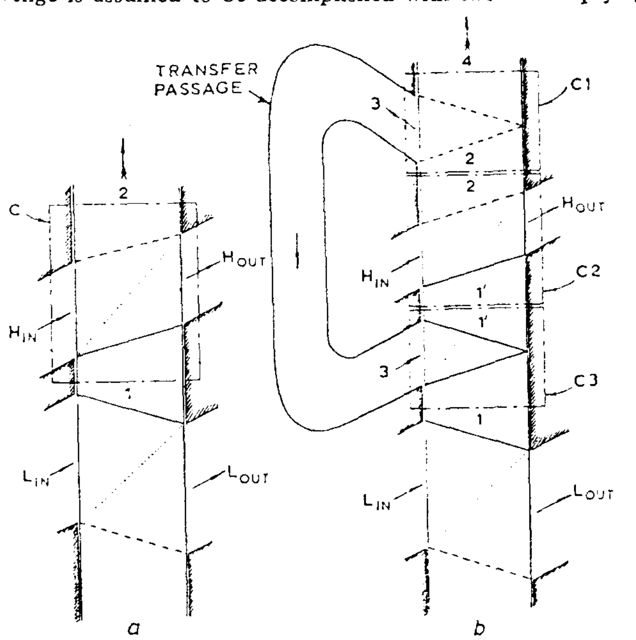


Fig. 29. Flow diagrams of DPE with and without transfer passage

and two cell-filling processes (i.e. with pre-compression) for all pressure ratios. Also, the port stagnation pressures at inlet and outlet in the low-pressure and high-pressure scavenge sections are assumed to be equal, i.e. $p_{H_{in}} = p_{H_{out}} = p_H$ and $p_{L_{in}} = p_{L_{out}} = p_L$. The high-pressure scavenge performance is taken to be represented by the following relation:

$$p_H/p_1 = f(p_2-p_1)/p_1 \quad . \quad . \quad . \quad (15)$$

where f is a certain function. (Note that in reversible pressure exchange, $\mu = 0$ and equation (10) reduces to $p_1 = p_2$.) The iterative method mentioned above yields the high-pressure scavenge performance curves shown in Fig. 30*a*.

The DPE overall pressure ratio p_H/p_L depends partly on the pre-compression in the low-pressure scavenge section, given by p_1/p_L, which in turn depends on the low-pressure scavenge performance. Let the overall performance be represented by the relation

$$p_H/p_L = f'(p_H-p_1)/p_L \quad . \quad . \quad . \quad . \quad (16)$$

where f' is a certain function. An iterative method similar to that used for high-pressure scavenge yields the overall performance curves of Fig. 30*b*.

If, in a qualitative first approach, it is assumed that the curves of Fig. 30 represent linear functions,

$$p_H/p_1 = 1+k(p_2-p_1)/p_1 \quad . \quad . \quad . \quad (17)$$

and

$$p_H/p_L = 1+c(p_H-p_1)/p_L \quad . \quad . \quad . \quad (18)$$

where c and k are constants (greater than unity) for any given value of δ. Equations (14), (17) and (18) then yield the following simplified general DPE performance equation:

$$p_H/p_L = 1+ck\gamma\mu \quad . \quad . \quad . \quad . \quad (19)$$

The generalized performance curves (shown in full lines) of Fig. 9 are based on the performance characteristics given in Fig. 30 and on the application of equation (14).

In the case of a SSPE, $p_1 \simeq p_L$ and $p_2 \simeq p_H$ so that equation (14) then approximates to

$$p_H/p_L = 1+\gamma\mu \quad . \quad . \quad . \quad . \quad (20)$$

GENERALIZED PERFORMANCE FOR A DPE WITH ONE TRANSFER PASSAGE

Since a transfer passage connects a high-pressure region to a low-pressure region of the rotor cells, the relevant aerodynamic processes are cell-emptying, from the rotor cells into one end of the passage, and cell-filling, from the other end of the passage into the rotor cells (see Fig. 29*b*). The stagnation pressures at both ends of the passage are taken to be equal. Application of the First Law of Thermodynamics to control surfaces c1 and c3 yields

$$c_v\dot{m}_2T_2 = c_p\dot{m}_3T_3+c_v\dot{m}_4T_4 \quad . \quad . \quad . \quad (21)$$

and

$$c_v\dot{m}_1T_1 = -c_p\dot{m}_3T_3+c_v\dot{m}_{1'}T_{1'} \quad . \quad . \quad (22)$$

Combining equations (21) and (22),

$$\dot{m}_1T_1+\dot{m}_2T_2 = \dot{m}_{1'}T_{1'}+\dot{m}_4T_4 \quad . \quad . \quad (23)$$

or

$$p_1+p_2 = p_{1'}+p_4 \quad . \quad . \quad . \quad . \quad (24)$$

A relation similar to equation (14) is obtained by applying the First Law of Thermodynamics to control surface c2:

$$\gamma\mu = (p_2-p_{1'})/p_L \quad . \quad . \quad . \quad . \quad (25)$$

μ is correlated to the DPE overall pressure ratio p_H/p_L by following a procedure similar to that outlined for the case of no transfer passage, as well as by applying equation (24). The high-pressure scavenge performance curves shown in Fig. 31*a* are identical with those shown in Fig. 30*a*. The overall performance curves (Fig. 31*b*), however, reveal a marked improvement in performance as compared with the corresponding performance for no transfer passage. If, for simplicity, the curves are assumed to represent linear functions, then

$$p_H/p_{1'} = 1+k(p_2-p_{1'})/p_1 \quad . \quad . \quad . \quad (26)$$

and

$$p_H/p_L = 1+c'(p_H-p_{1'})/p_L \quad . \quad . \quad . \quad (27)$$

where k is the same constant as that in equation (17) and c' is estimated from Figs 30*b* and 31*b* to be of the order of 2·6c for all values of δ. Equations (25), (26) and (27) yield the following simplified general performance equation for a DPE with one transfer passage:

$$p_H/p_L = 1+c'k\gamma\mu \quad . \quad . \quad . \quad . \quad (28)$$

The generalized performance curves (shown in full lines) of Fig. 10 are based on the performance characteristics shown in Fig. 31 and on the application of equation (25).

In the case of a SSPE, $p_1 \simeq p_L$ and $p_{1'} \simeq p_4$ so that equation (24) becomes

$$p_L+p_H = 2p_{1'} \quad . \quad . \quad . \quad . \quad (29)$$

Also, since $p_2 \simeq p_H$, equation (25) may be re-written as

$$p_H/p_L = 1+2\gamma\mu \quad . \quad . \quad . \quad . \quad (30)$$

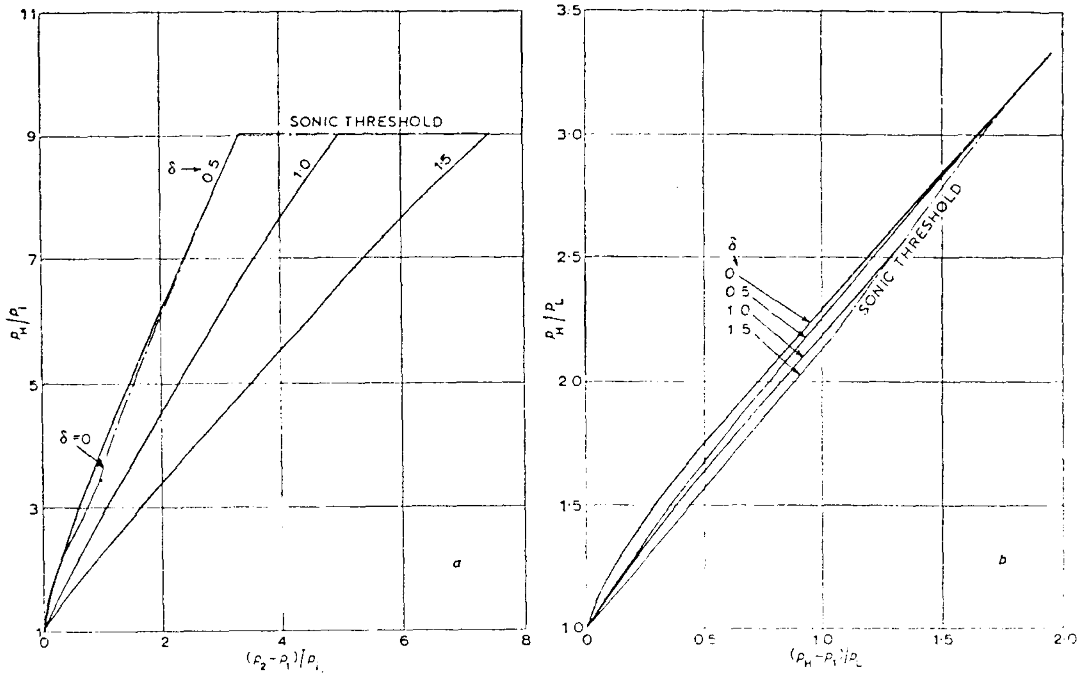


Fig. 30. Scavenge performance for a DPE with no transfer passage

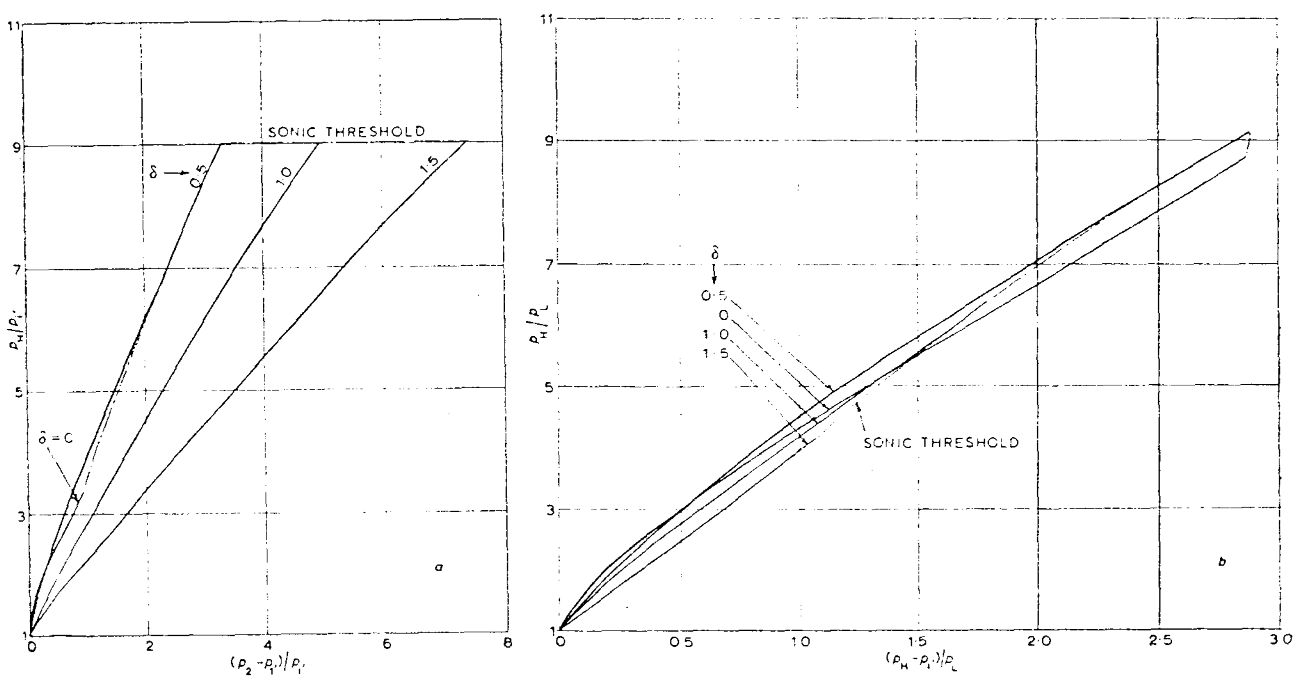


Fig. 31. Scavenge performance for a DPE with one transfer passage

APPENDIX II

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Journal of Basic Engineering/Transactions of the ASME
September 1969, p. 361-370

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The Performance of Pressure-Exchanger Dividers and Equalizers

Tests are reported which were carried out to establish the performance of pressure-exchanger dividers and equalizers using air as the working fluid. The effects of varying rotor speed, port geometry, and port pressure ratios were explored. It was concluded that, at low pressure ratios, pressure-exchanger dividers and equalizers are comparable in performance with their turbomachine counterparts.

Introduction

RESULTS of tests carried out on pressure-exchang intended for use as internal combustion engine superchargers etc., employing gas-generator-type cycles have been reported in the literature [1, 2, 3].[1] In this paper a presentation is made of experimentally obtained results for two additional pressure-exchanger cycles suitable for other applications; namely, the pressure divider and pressure equalizer cycles.

A divider is a machine in which an entering gas stream is split (or divided) into two outgoing flows, one having a higher and the other a lower stagnation pressure than the ingoing flow. An equalizer is, in essence, a divider with the flow direction of all three streams reversed. More detailed descriptions of pressure-exchanger divider and equalizer cycles have been given by Azoury [4] who also compared these cycles with those of analogous turbomachines.

A divider can be used as a pressure booster for which application the useful output is the high pressure stream. Attention has also been drawn to the possibility of using the low pressure stream leaving a divider as a low temperature sink for cooling purposes [4, 5, 6]. An equalizer performs a similar function to an ejector or a turbocompressor unit in which the flows leaving the turbine and the compressor are at the same pressure [7].

The prime reasons for choosing pressure-exchanger dividers or equalizers in preference to their turbomachine counterparts are to enable the benefits to be derived from the robust construction and low rotational speed of the former. Naturally, such choices will be strongly influenced by the relative merits of the performances of the pressure-exchangers.

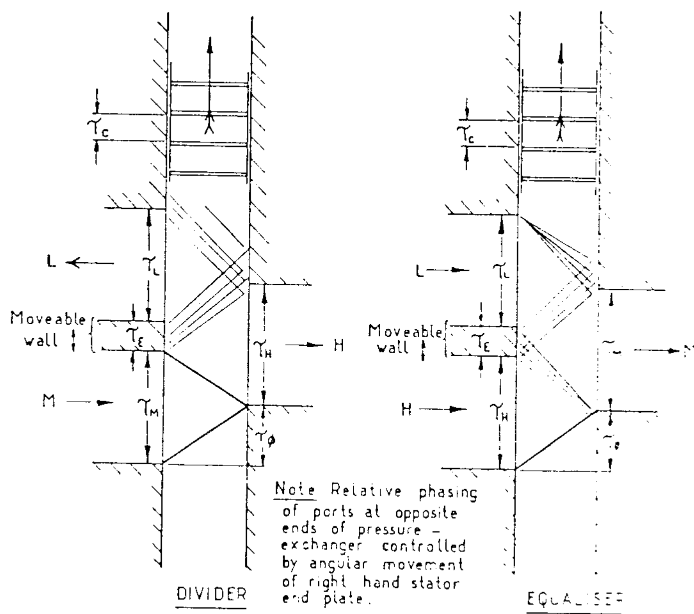

Fig. 1 Divider and equalizer port arrangements

In the program of work reported here the performances of a pressure-exchanger divider, operating as a pressure booster, and an equalizer were investigated experimentally using air as the working fluid. The tests were carried out on a dual-purpose pressure-exchanger rig operable as a divider or equalizer. It was essential that the particular port arrangements chosen were such that conversion from one role to the other could be carried out at minimum cost. Fig. 1 shows the basic forms of divider and equalizer on which the tests were carried out. It can be seen from Fig. 1 that the only reversal in flow direction resulting from conversion was in the low pressure (LP) ports, the divider LP outlet corresponds to the equalizer LP inlet. Alternative divider and equalizer port arrangements to those chosen are possible [6].

[1] Numbers in brackets designate References at end of paper.

Contributed by the Fluid Mechanics Committee and presented at the Winter Annual Meeting, New York, N. Y., December 1–5, 1968, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. Manuscript received at ASME Headquarters, August 19, 1968. Paper No. 68—WA/FE-24.

Nomenclature

a_0 = velocity of sound at stagnation conditions (subscript identifying the flow is placed to the left of the "0")

a_{ref} = reference velocity of sound

A_{port} = port area

c_p = specific heat at constant pressure

$\dot{m}$ = mass flow

M = axial component of Mach number

M_{R_0} = dimensionless speed of rotor at mean radius of cells, $\equiv \dfrac{u_w}{a_{ref}}$

N = speed of rotor, rpm

P = static pressure

P_0 = stagnation pressure; the subscript identifying the flow is placed to the left of the "0"

T_0 = stagnation temperature; the subscript identifying the flow is placed to the left of the "0"

u_w = circumferential velocity of rotor at mean radius of cells

β = divider mass ratio, $\equiv \dot{m}_H/\dot{m}_M$

γ = ratio of specific heats, $\equiv c_p/c_v$

ζ = equalizer performance parameter, $\equiv \dot{m}_L T_{L0}/\dot{m}_H T_{H0}$

η = divider or equalizer overall isentropic efficiency

ρ = fluid density based on average static conditions

τ = dimensionless time, $\equiv$ (time $\times$ a_{ref})/cell length

τ_c = width of cell in dimensionless time units

τ_ϵ = width, in dimensionless time units, of land between adjacent ports

τ_φ = dimensionless time equivalent of port phasing angle

Subscripts

H, L, M = high, low, or medium pressure flow, respectively

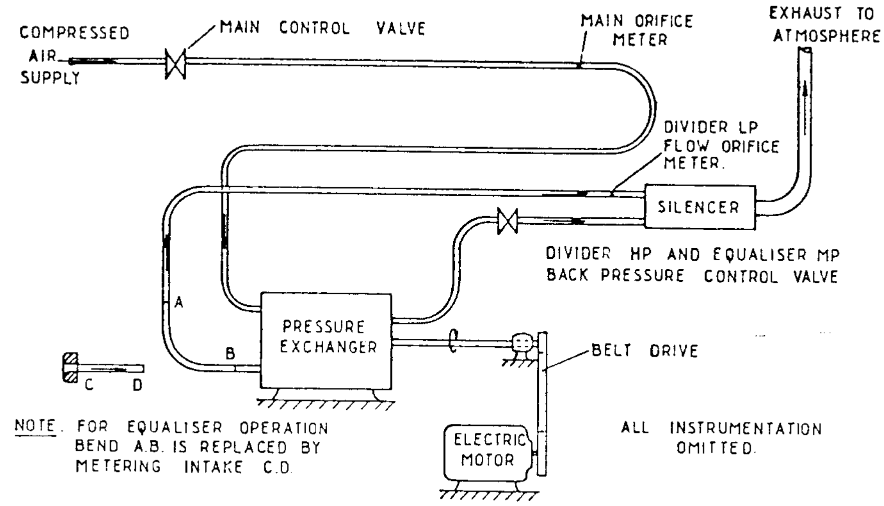

Fig. 2 Pressure-exchanger test rig circuit diagram

The rig pressure-exchanger was equipped with variable port geometry and with a variable speed drive to the rotor. It was found convenient to obtain a set of experimental results for each geometry, at a fixed rotor speed, presenting the performance on a plane the abscissa and ordinate of which were stagnation pressure ratios [4, 6]. For the divider lines of constant β ($\equiv \dot{m}_H/\dot{m}_M$) and lines of constant Mach number ($\mathbf{M}_M$) in the inlet ports were drawn on a P_{L0}/P_{M0} versus P_{H0}/P_{M0} plane. The corresponding presentation for the equalizer consisted of lines of constant ζ ($\equiv \dot{m}_L T_{L0}/\dot{m}_H T_{H0}$), and others of constant Mach number ($\mathbf{M}_H$) in the high pressure inlet ports, on a P_{H0}/P_{M0} versus P_{L0}/P_{M0} plane. It is a particular advantage of the chosen planes that constant overall pressure ratio (P_{H0}/P_{L0}) is represented by families of inclined straight lines. It has been shown elsewhere [6] that the presentations contained sufficient information to represent adequately the performances obtained in the experiments.

Test Rig

The pressure-exchanger used for the experiments was designed by the sponsors of the project, Power Jets (Research & Development) Ltd., and was specially conceived for testing divider and equalizer cycles. The design centered around a rotor of orthodox pressure-exchanger type. The rotor, which contained 30 axially disposed cells, was 11 in. in length and 8 in. in dia. The clearance between the rotor and each end face of the stator was approximately 0.007 in. Provision was made to allow for relative angular movement of the stator end faces, and also for altering the relative port sizes by means of built-in variable width ducts at one end of the machine, as shown in Fig. 1. The rotor was driven by a variable speed d-c motor by way of a toothed belt drive and countershaft, the nominal design speed of the rotor was 6000 rpm. The electric drive insured that the operator was always in control of rotor speed.

In order to make best use of the stator end plates three complete sets of high, medium, and low pressure ports were provided; six ports were, therefore, situated at one end of the machine, the remaining three at the opposite end. A photograph of the pressure-exchanger rig set up for divider tests is given by Azoury [4]. Fig. 2 shows the rig circuit.

The air mass flow supplied to the pressure-exchanger was measured by means of an orifice meter. An orifice meter was also used to measure the flow in the divider LP outlet duct, flow leaving the duct was discharged into the atmosphere. The orifice meter was installed in the LP outlet duct in preference to the high pressure (HP) outlet because of the presence of the back pressure throttling valve in the latter, Fig. 2. The equalizer LP inlet flow was measured by means of a bell mouth meter attached to the upstream end of a short inlet pipe. The bell

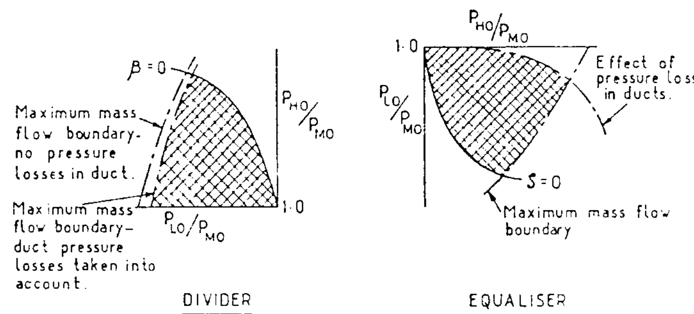

Fig. 3 Limitations of the explorable regions in the divider and equalizer performance tests

mouth also served to minimize the loss of pressure sustained by the LP stream drawn from the surroundings. Pressure and temperature measurements contributing to the test data were made in the ports and ducts. Substantially steady flow prevailed everywhere external to the cells so these measurements, which were made by conventional steady flow techniques, did not present any major difficulties.

Limitations were imposed on the performance of the rig by the maximum compressed air mass flow available and by pressure losses in the ducts. For the divider arrangement the most serious pressure loss was that in the LP outlet duct. This loss had the effect of elevating the pressure level in the LP outlet ports and, as a consequence, it shifted the maximum mass flow performance boundary further to the right on the P_{L0}/P_{M0} versus P_{H0}/P_{M0} plane than would otherwise have been the case. The range of the equalizer performance field at $P_{L0}/P_{M0} \rightarrow 1$ was limited by the pressure loss in the medium pressure (MP) outlet duct, the magnitude of this effect increased with increasing mass flow. The explorable regions, on the divider and equalizer performance presentation planes are shown qualitatively as cross-hatched areas in Fig. 3.

It appears that the pressure-loss restrictions on the performance of the rig could have been relaxed by fitting larger ductwork, etc. The 6-in. ducts used on the rig were, however, about the largest permissible within the compass of space and cost limitations.

Results of Experiments

In order to generalize the experimental results and to assist their comparison with theoretical performance predictions, rotor speed, port widths, phasing angles, etc., have been expressed in dimensionless units. The dimensioned forms directly applicable to the Power Jets machine are also displayed on the presentations of the experimental performance results.

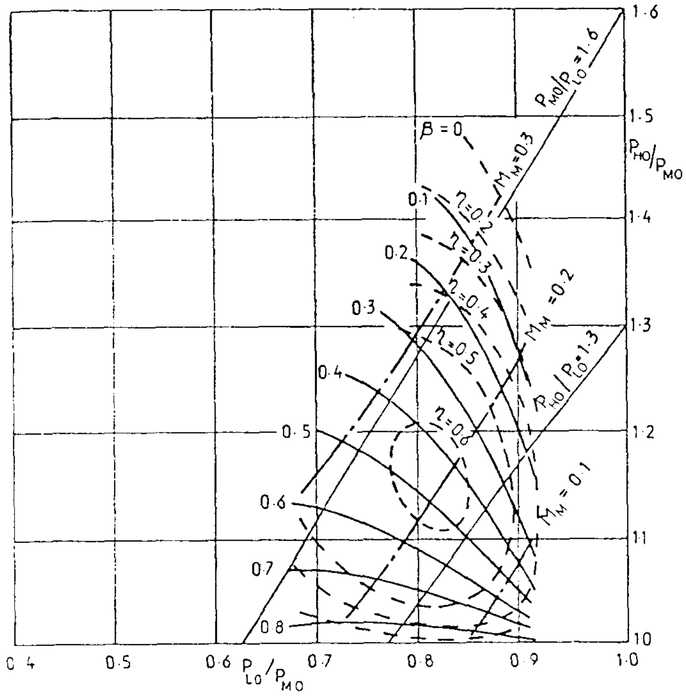

Fig. 4 Divider performance map, fixed geometry, M_{R0} = 0.147 (6000 rpm)

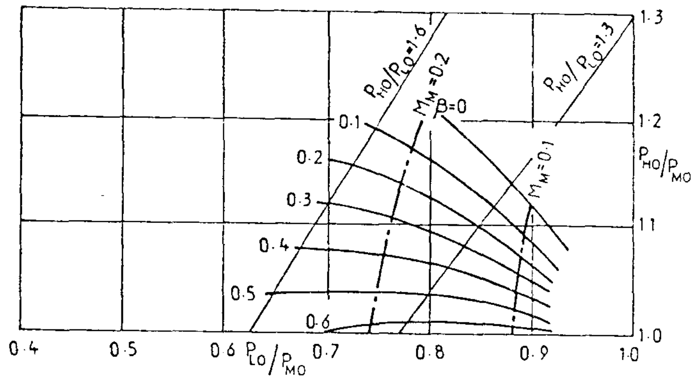

Fig. 5 Divider performance map, fixed geometry, M_{R0} = 0.098 (4000 rpm)

Because the divider MP and equalizer HP inlet stagnation temperatures were maintained at a unique value (±3 percent) throughout the whole series of experiments, it was therefore convenient to use that temperature as the basis for the (constant) reference sonic velocity in the dimensionless forms for rotor speed and port geometry. Dimensionless rotor speed, M_{R0}, is defined by:

$$M_{R0} \equiv \frac{u_\omega}{a_{ref}} \begin{cases} \text{where } a_{ref} \equiv a_{M0} \text{ for dividers,} \\ \phantom{\text{where }} a_{ref} \equiv a_{H0} \text{ for equalizers.} \end{cases}$$

Note: For all experiments $a_{M0} \approx a_{H0} \approx 1155$ fps.

Port widths, phasing angle, cell width, etc., are expressed in terms of dimensionless time τ. A τ value of unity is the dimensionless time taken for an accoustic wave to travel the length of a cell at the stagnation sonic velocity a_{ref}. A port width in terms of τ is therefore the dimensionless time required for a point on the rotor to pass the port opening; this is clearly a function of rotor speed. Another consequence of the constancy of the reference temperature was that dimensioned speed (N rpm) could be used as a performance parameter instead of the $N\sqrt{T_0}$ form.

A problem arose in the interpretation of the stagnation pressures in outlet ports. Appreciable nonuniformities of velocity, and therefore of stagnation pressure, were found to exist, the nature of which depended upon the port geometry and test conditions. Such nonuniformities are well known in pressure-exchangers and are caused by wave events within the rotor; they are generally predictable by the method-of-characteristics [4]. Accordingly, the stagnation pressure in each outlet port was established from knowledge of the mass flow and the average port static pressure by substitution in the well-known isentropic relation:

$$P_0 = P\left[1 + \left(\frac{\gamma - 1}{2}\right) \mathsf{M}^2\right]^{\left(\frac{\gamma}{\gamma - 1}\right)} \tag{1}$$

where P is the average static pressure in the port.

M is the average Mach number based on the port mass flow and average static conditions; thus,

$$P_0 = P^{\frac{-1}{\gamma - 1}}\left[P + \left(\frac{\gamma - 1}{2\gamma}\right) \frac{\dot{m}^2}{\rho(A_{\text{port}})^2}\right]^{\left(\frac{\gamma}{\gamma - 1}\right)} \tag{2}$$

It may be shown that P_0 obtained in this way is a close approximation of the value which would prevail, at a station across which the conditions are uniform, if the port momentum and mass flow are conserved in a frictionless duct of the same cross-sectional area as the port [6].

Divider. Because each test was carried out with a fixed geometry and at a constant speed, each set of results was plotted on the P_{L0}/P_{M0} versus P_{H0}/P_{M0} plane. Pictures of the effects on performance of changes of geometry or speed were, therefore, compiled by cross-plotting from the P_{L0}/P_{M0} versus P_{H0}/P_{M0} presentations. Figs. 4 and 5 are typical of performance presentations for fixed geometry and constant speed.

The effect of changes of τ_φ on performance when port width ratio and rotor speed were maintained constant at M_{R0} = 0.147 (6000 rpm) is shown in Figs. 6 and 7. The criterion of merit on the diagrams, which were each drawn for a specified overall pressure ratio P_{H0}/P_{L0}, is the maximum P_{H0}/P_{M0} attainable with a given value of β. It is, therefore, clear that the best performance was obtained with $\tau_M = \tau_L = 1.57$. The optimum phasing showed a tendency to decrease as P_{H0}/P_{L0} was increased, optimum τ_φ being about 0.94 when P_{H0}/P_{L0} = 1.3, and 0.84 when P_{H0}/P_{L0} = 1.6. A tendency for τ_φ to increase as β decreased was also slightly in evidence. The absence of any great sensitivity in the effect on performance of small changes in either port widths or phasing is noteworthy.

Fig. 4 shows the best (fixed geometry) divider performance obtained from the experiments. The medium and low pressure widths were $\tau_M = \tau_L = 1.57$, the phasing was $\tau_\varphi = 0.84$. Fig. 4 displays, in addition to contours of β and M_M, lines of constant η. These were evaluated from the relationship:

$$\eta = \left(\frac{\beta}{1 - \beta}\right) \frac{\left[(P_{H0}/P_{M0})^{\frac{\gamma - 1}{\gamma}} - 1\right]}{\left[1 - (P_{L0}/P_{M0})^{\frac{\gamma - 1}{\gamma}}\right]} \tag{3}$$

η is the equivalent of the product of the isentropic efficiencies of compression and expansion in a turbomachine analogy of the divider [4]. Therefore η is a criterion of merit of the performance of the divider, a criterion which allows the divider to be compared directly with a turbomachine alternative. A derivation of equation (3) will be found elsewhere [6].

Figs. 8 and 9 present, over a range of rotor speeds, the performance of the machine at what was found to be the optimum (dimensioned) geometry at M_{R0} = 0.147 (6000 rpm), that is with a phasing angle = 24 deg, MP = LP port width = 45 deg, HP port width = 48 deg. Because port widths, etc., expressed in terms of dimensionless time are directly proportional to the angle subtended, and inversely proportional to rotor speed, the dimensionless geometry applicable to Figs. 8 and 9 is not independent of rotor speed; curves of dimensionless geometrical properties are superimposed in Fig. 8.

It can be seen from Figs. 8 and 9 that the performance alters

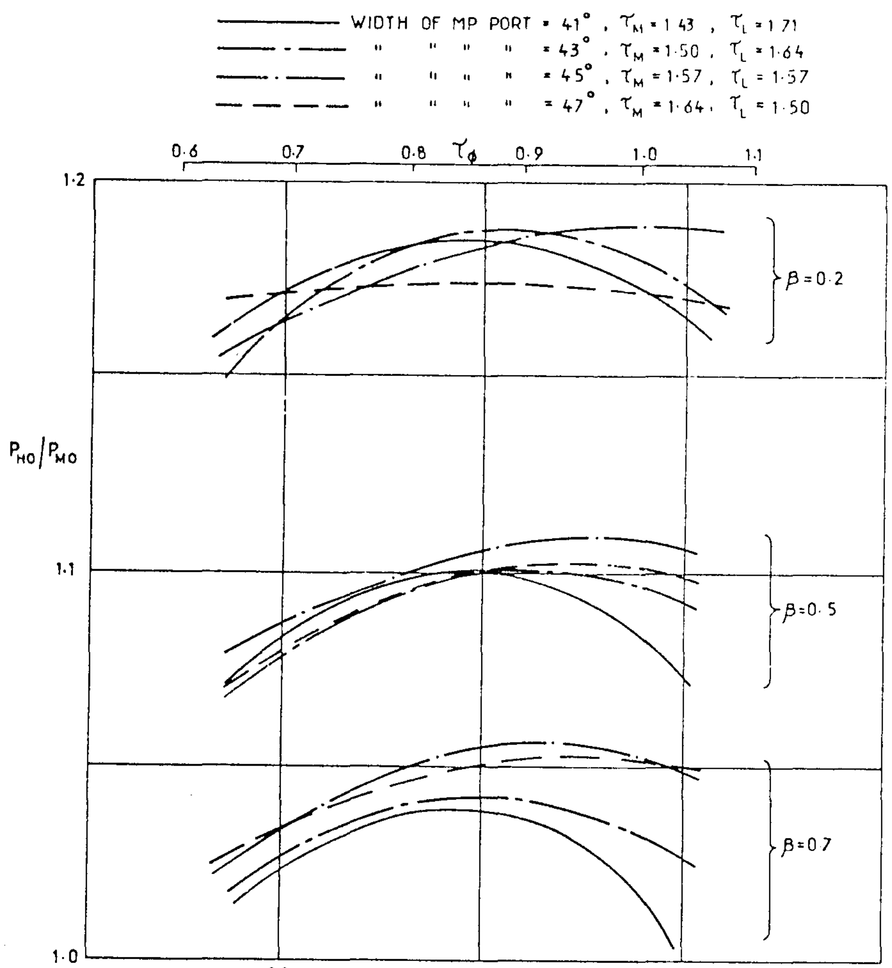

Fig. 6 Divider performance variation with port geometry, P_{H0}/P_{L0} = 1.3, M_{R0} = 0.147 (6000 rpm)

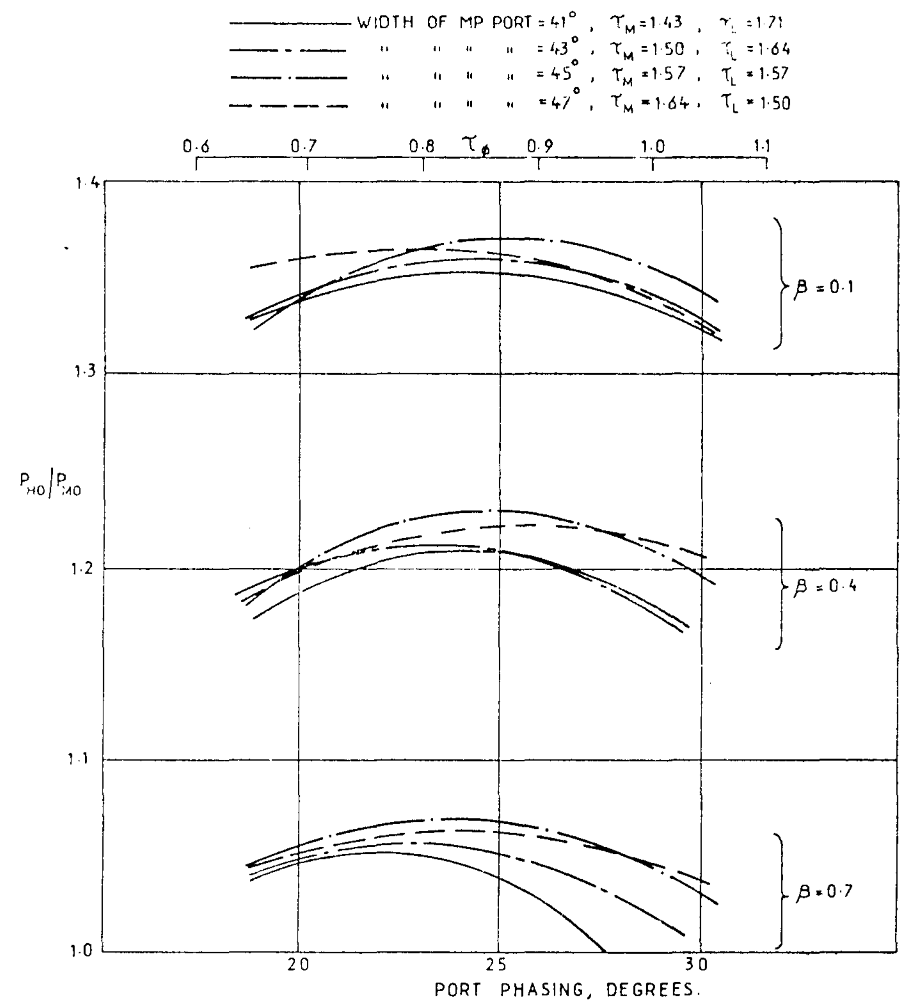

Fig. 7 Divider performance variation with port geometry, P_{H0}/P_{L0} = 1.6, M_{R0} = 0.147 (6000 rpm)

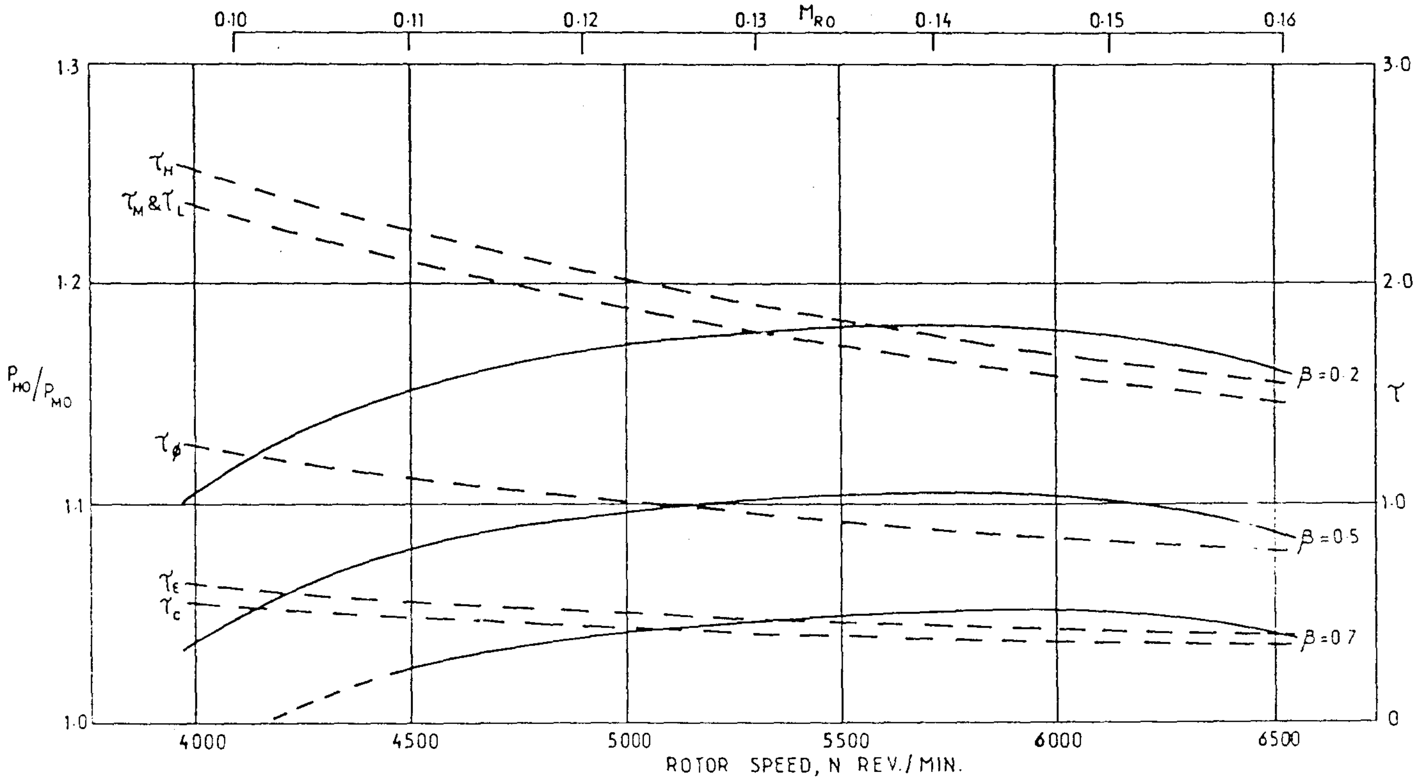

Fig. 8 Divider performance variation with rotor speed, fixed geometry, P_{H0}/P_{L0} = 1.3

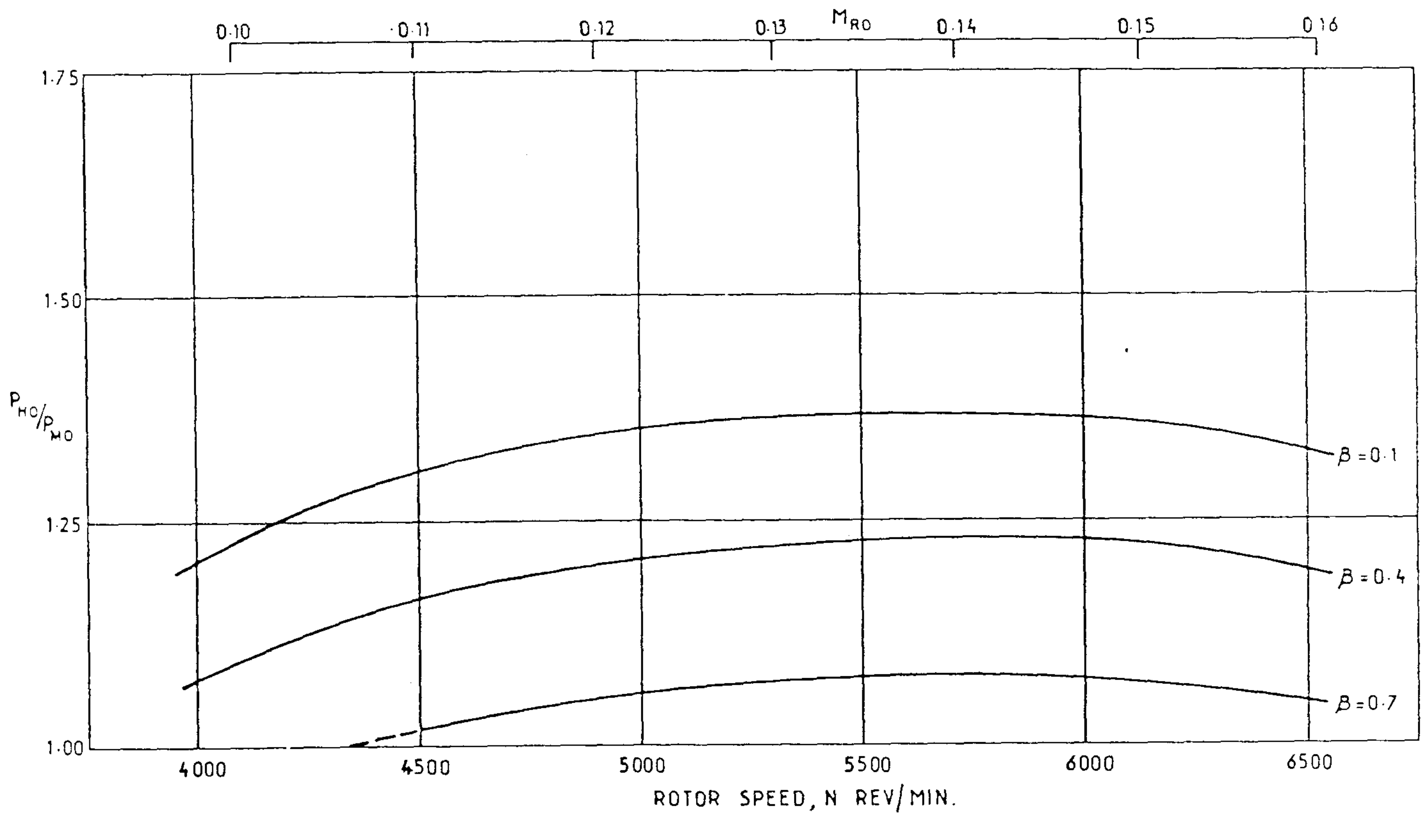

Fig. 9 Divider performance variation with rotor speed, fixed geometry, P_{H0}/P_{L0} = 1.6

but little between M_{R0} = 0.135 (5500 rpm) and M_{R0} = 0.147 (6000 rpm), and worsens as the speed is raised above or reduced below these limits. At M_{R0} = 0.098 (4000 rpm) the performance had depreciated to such an extent that the pressure rise, for a given value of β, was only about half that obtained when M_{R0} = 0.147. This point is clarified by comparing Fig. 5, the performance on the P_{L0}/P_{M0} versus P_{H0}/P_{M0} plane at M_{R0} = 0.098, with Fig. 4.

The accuracy of the divider experiments was estimated to correspond, in effect, to a maximum uncertainty in the value of β of approximately ± 0.04. Uncertainties of lesser magnitude prevailed for values of β approaching zero and approaching unity.

Equalizer. The results of each complete test to determine equalizer performance characteristics with fixed geometry at a constant speed, were displayed on the P_{H0}/P_{M0} versus P_{L0}/P_{M0} plane; Figs. 10 and 11 are examples of this presentation. As for

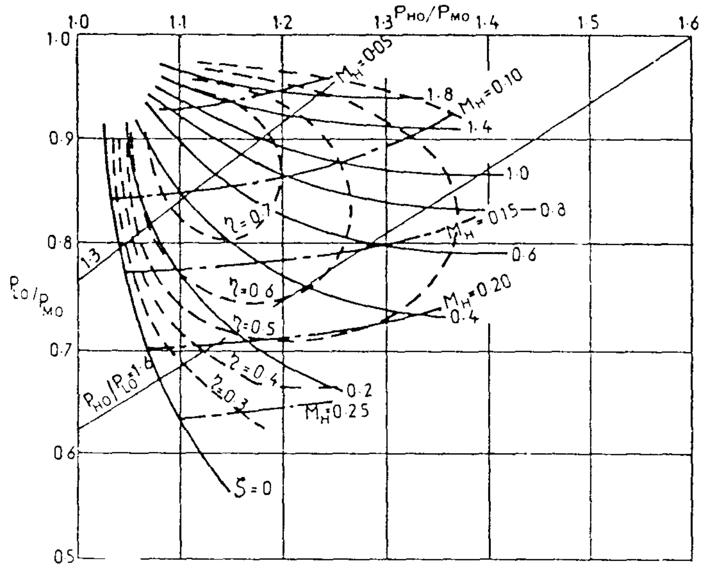

Fig. 10 E[illegible]alizer performance map, fixed geometry, M_{R0} = 0.135 (5500 rpm)

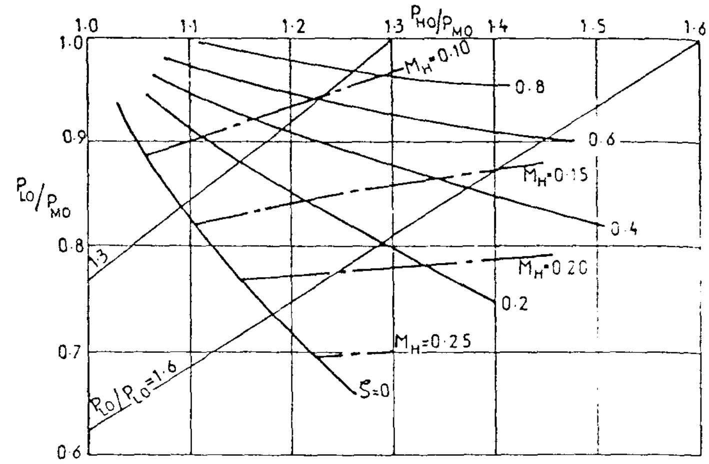

Fig. 11 Equalizer performance map, fixed geometry, M_{R0} = 0.098 (4000 rpm)

the divider, the effects on performance of changes of port geometry or rotor speed were determined by cross-plotting from the fixed geometry, constant speed, experimental results.

Figs. 12 and 13 show the effect on performance of variation of phasing τ_φ, when the overall pressure ratio P_{H0}/P_{L0} and rotor speed are maintained constant. For Fig. 12 $P_{H0}/P_{L0} = 1.3$, for Fig. 13 $P_{H0}/P_{L0} = 1.6$; for both cases the rotor speed corresponded to $M_{R0} = 0.135$ (5500 rpm). Separate curves are shown on each diagram for HP port widths ranging from $\tau_H = 1.48$ to $\tau_H = 1.94$. The best geometrical arrangement for any prescribed ζ, and of course overall pressure ratio, is that which produced the lowest P_{L0}/P_{M0}.

From an examination of Fig. 12 it can be seen that the best port width is $\tau_H = 1.94$ with $\tau_\varphi = 0.95$ when $\zeta = 0$, changing to $\tau_H = 1.79$ with $\tau_\varphi = 1.00$ when $\zeta = 1.0$. Fig. 13 shows that the best performance is obtained when $\tau_H = 1.94$ with $\tau_\varphi = 0.95$ at $\zeta = 0$, changing to $\tau_H = 1.48$ with $\tau_\varphi = 0.70$ at $\zeta = 1$.

It was concluded from Figs. 12 and 13, that, for $M_{R0} = 0.135$, very little loss of performance was incurred over the range of pressure ratios explorable, if the geometry were fixed with an HP port width given by $\tau_H = 1.79$, when $\tau_\varphi = 0.91$. The performance with this geometry for $M_{R0} = 0.135$ is shown on the P_{H0}/P_{M0} versus P_{L0}/P_{M0} plane in Fig. 10. This was the best performance, at fixed speed and geometry, revealed by the equalizer tests. Contours of isentropic product efficiency η have been added to the diagram; it may be noticed that the maximum efficiency occurs fairly close to the $P_{H0}/P_{M0} = P_{L0}/P_{M0} = 1$ origin. The isentropic product efficiency serves a similar purpose; namely, in permitting direct comparison with a turbomachine analogy, for both equalizers and dividers. For equalizers η is given by [6]:

$$\eta = \zeta \frac{\left[(P_{M0}/P_{L0})^{\frac{\gamma-1}{\gamma}} - 1\right]}{\left[1 - (P_{M0}/P_{H0})^{\frac{\gamma-1}{\gamma}}\right]}. \qquad (4)$$

Equation (4) is restricted to cases in which γ and c_p are the same for both the primary and secondary flows.

Figs. 14 and 15 show the performance of the equalizer arrangement over a range of rotor speeds between $M_{R0} = 0.10$ and $M_{R0} = 0.15$, the latter being the highest speed attainable with the Power Jets machine working as an equalizer. The overall pressure ratios applicable to Figs. 14 and 15 are $P_{H0}/P_{L0} = 1.3$ and $P_{H0}/P_{L0} = 1.6$, respectively. Both diagrams were constructed for a fixed geometrical arrangement giving optimum performance in the $M_{R0} = 0.123$ to 0.135 speed range, that is, with an HP port width = 47 deg, LP port width = 43 deg, MP port width = 48 deg, with a phasing angle of 24 deg. The corresponding dimensionless form of this geometry has been plotted, over the appropriate speed range in Fig. 14.

From Figs. 14 and 15 it can be seen that the best rotor speed is approximately $M_{R0} = 0.13$. There is a slight tendency, noticeable in both diagrams, for the optimum speed to increase slightly as ζ increases.

Fig. 11 shows the performance, on the P_{H0}/P_{M0} versus P_{L0}/P_{M0} plane, at the low speed extremity of the curves shown in Figs. 14 and 15. Fig. 10 displays the performance characteristics for $M_{R0} = 0.135$; this contributes some of the best results appearing in Figs. 14 and 15.

The estimated accuracy of the experimental results for the equalizer was such that it corresponded to a maximum uncertainty in ζ of approximately ±4 percent.

Concluding Remarks

The experimental results for both the divider pressure booster and equalizer cycles showed that the regions of greatest product efficiency were located at values of P_{H0}/P_{M0} and P_{L0}/P_{M0} fairly close to unity. These maxima, which were roughly comparable with values of product efficiency obtainable from turbomachines, were approximately $\eta = 0.6$ and $\eta = 0.7$ for the divider and equalizer, respectively. There was no real indication in any of the divider or equalizer experiments that the regions of maximum efficiency could be moved appreciably by changing any of the experimental variables. The experiments also showed that the best performances were not, in general, sensitive to small changes in speed and port geometry.

It can be concluded, therefore, that pressure-exchanger dividers and equalizers of the types tested experimentally are feasible for operation at low pressure ratios and, for this condition, the performances obtainable are competitive with those of turbomachines. The use of a pressure-exchanger allows advantage to be taken of its robust construction and low rotational speed.

More information could have been obtained from the experiments had it been possible to explore the effect of varying the width of the divider high pressure, or equalizer medium pressure outlet ports. The width chosen for these ports was, however, a reasonable compromise based on a method-of-characteristics analysis of the requirements of each cycle.

Theoretical analysis shows that a somewhat superior divider pressure boosting performance should be obtainable with an alternative port arrangement resulting in waves of reduced strength. Both dividers and equalizers can, in principle, be compounded to operate at higher pressure ratios than single-stage machines [6].

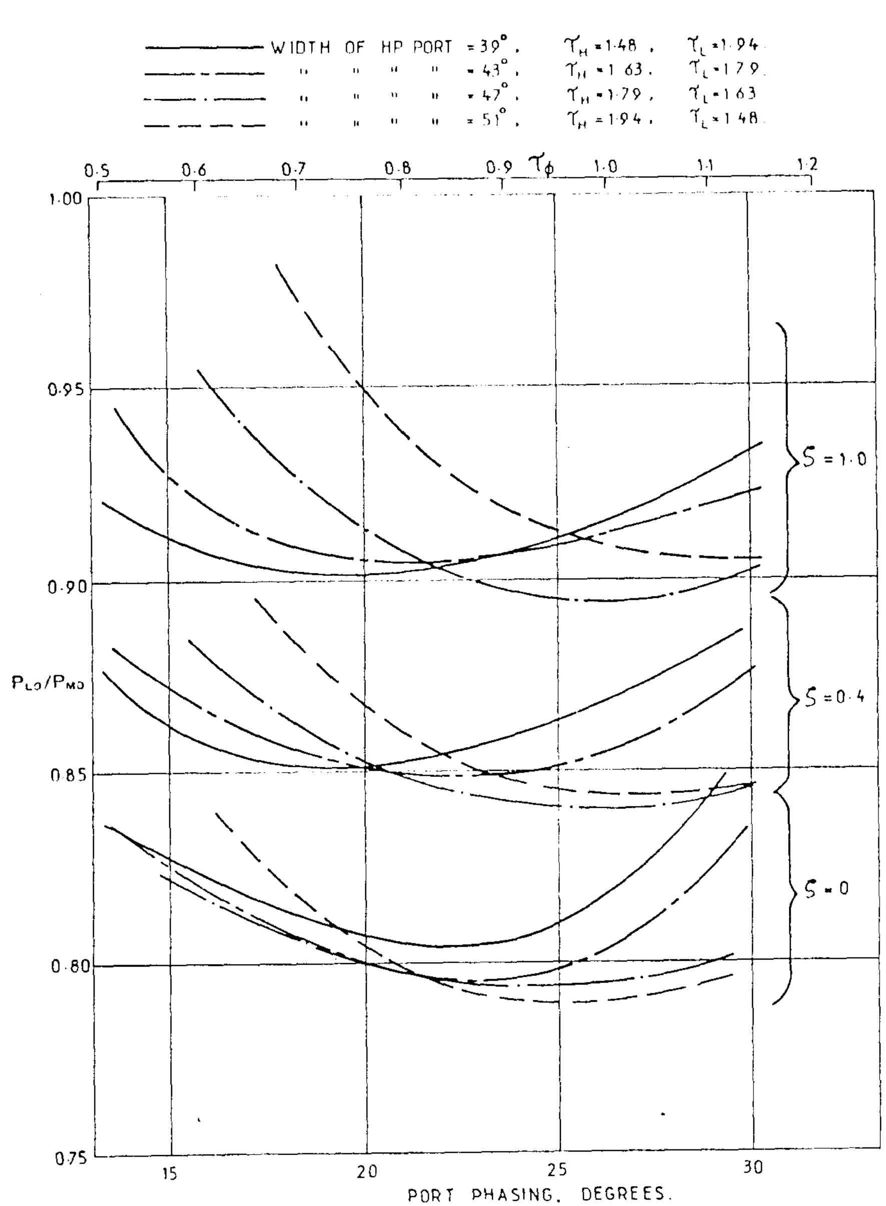

Fig. 12 Equalizer performance variation with port geometry, P_{HO}/P_{LO} = 1.3, M_{RO} = 0.135 (5500 rpm)

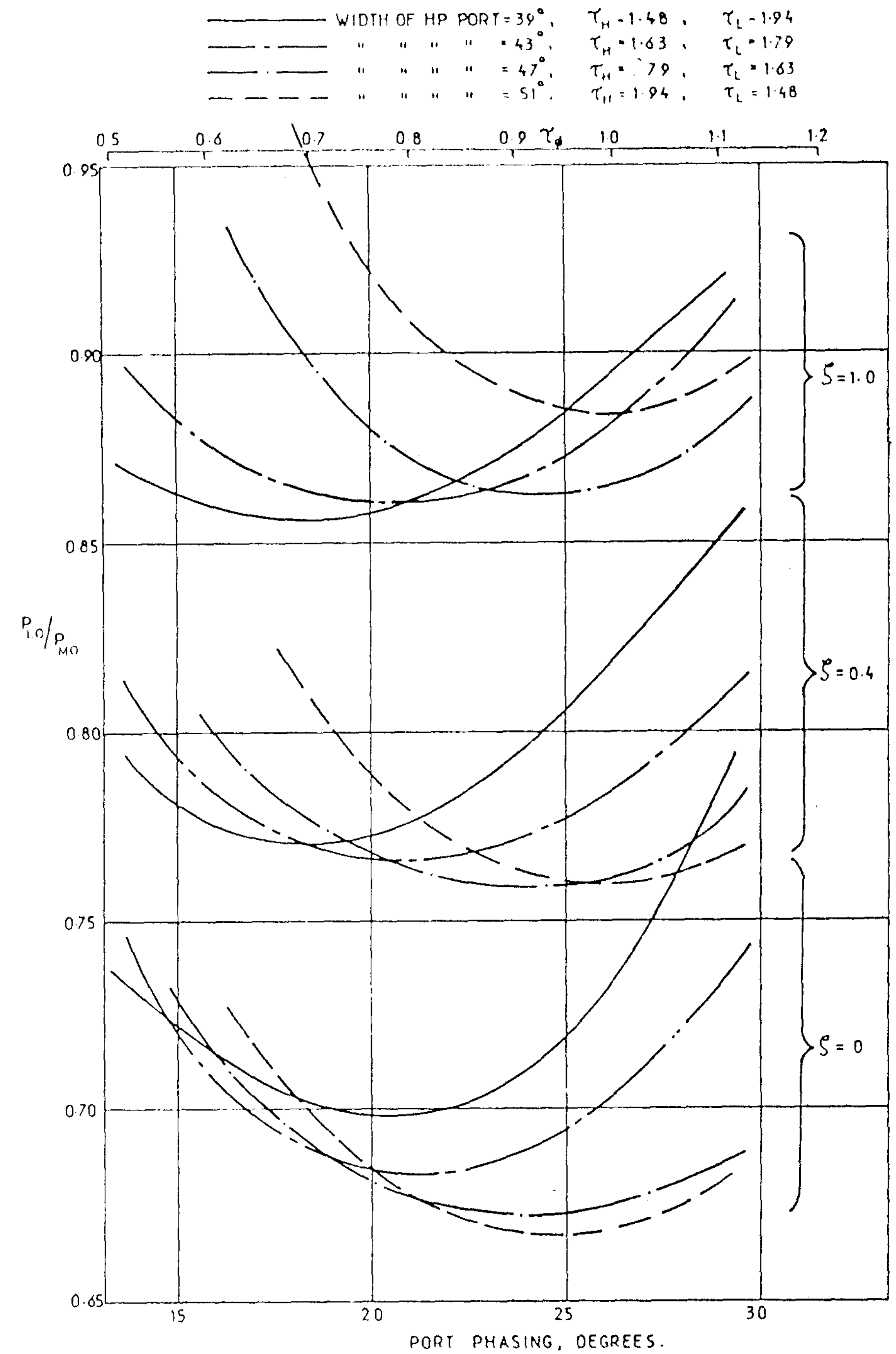

Fig. 13 Equalizer performance variation with port geometry, P_{HO}/P_{LO} = 1.6, M_{RO} = 0.135 (5500 rpm)

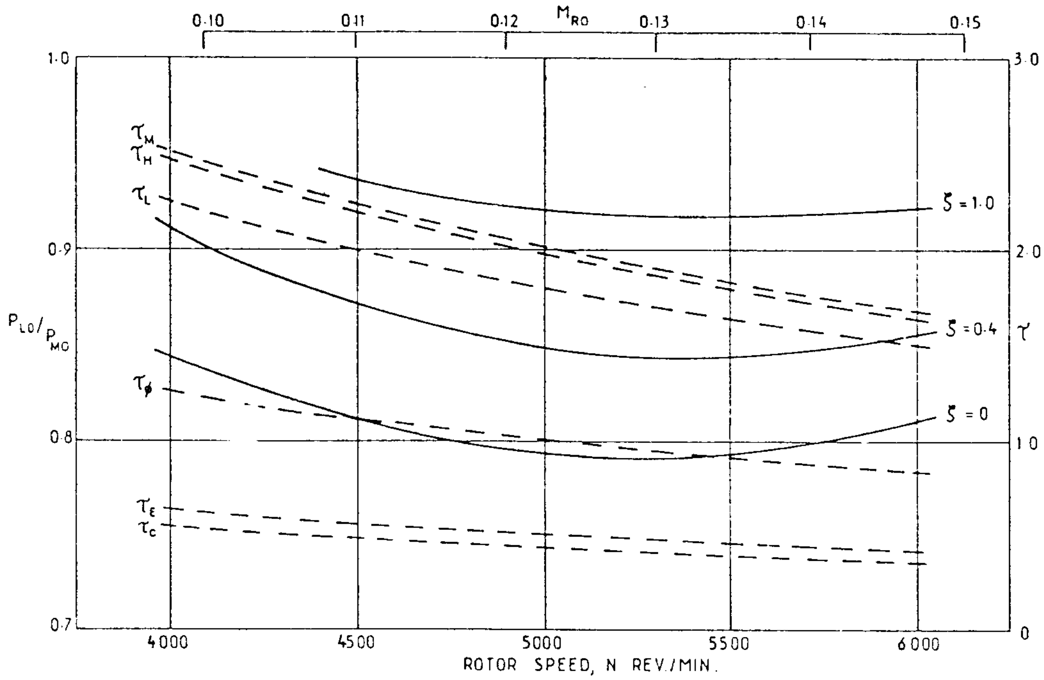

Fig. 14 Equalizer performance variation with rotor speed, fixed geometry, $P_{H0}/P_{L0} = 1.3$

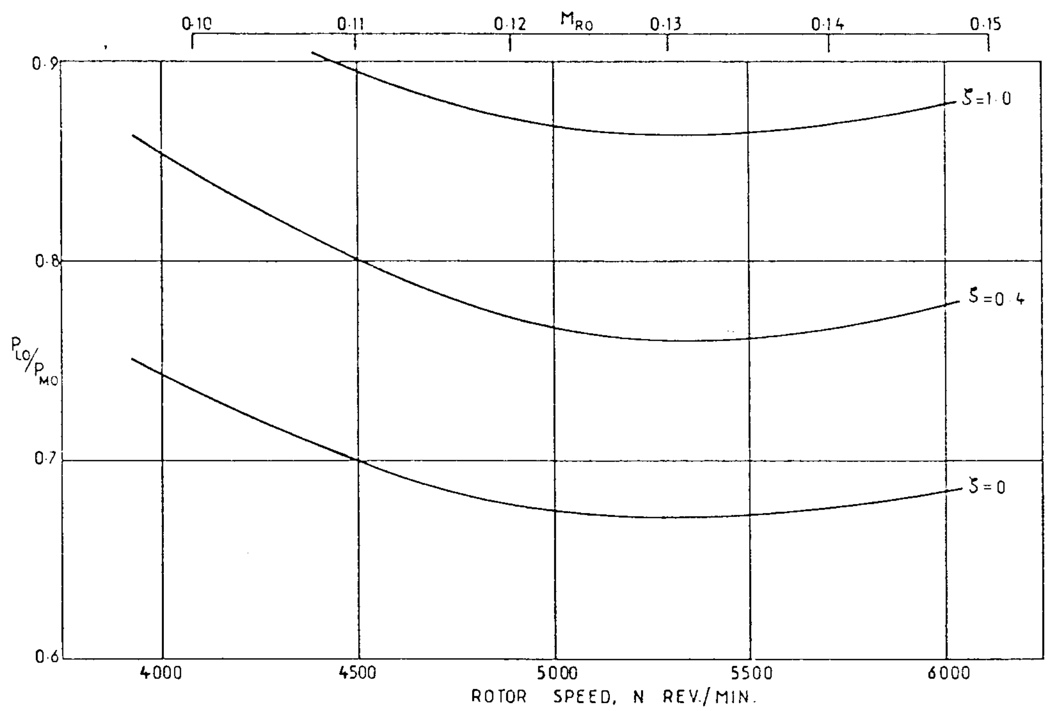

Fig. 15 Equalize performance variation with rotor speed, fixed geometry, $P_{H0}/P_{L0} = 1.6$

Acknowledgments

The writer wishes to acknowledge the help of Power Jets (R&D) Ltd. and Prof. D. B. Spalding of Imperial College for sponsoring and supervising the work, respectively.

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DISCUSSION

P. H. Azoury[2]

In recent years, there has been a steadily growing interest in pressure exchangers of the orthodox rotor type considered by Dr. Kentfield. His present contribution to the field is most welcome.

It is well known that, for optimum performance, simple single-stage pressure exchangers of the type just mentioned must operate at overall pressure ratios less than about 2.5. Dr. Kentfield's experimental analysis has shown that the most efficient running of pressure-exchanger dividers and equalizers occurs with overall pressure ratios of the order of 1.4. There is no doubt that these units, when compared with conventional turbomachines and steady-flow ejectors, can find applications where their erosion-resistance potentialities and/or their superior efficiency renders them highly attractive or even uniquely suitable. The lowness in overall pressure ratio, however, is restrictive and simple means must be found for displacing the region of maximum efficiency to higher working pressure ratios. Dr. Kentfield has mentioned compounding as one possible means and has suggested elsewhere [4] the use of feedback ducts as another. I would welcome Dr. Kentfield's comments on the relative merits of these two methods in the light of the potential commercial applications of the units.

Dr. Kentfield has already presented a performance estimate of the equalizer as a thrust augmenter using the efflux of a nonafterburning turbojet engine as a source of driving fluid, the driven fluid being air entrained from the atmosphere [4]. It appears, however, that a more promising application of the equalizer would be as a thrust augmenter for an afterburning turbofan, in which case it should be possible for the equalizer to benefit simultaneously from its higher efficiency, as compared to the steady-flow ejector, at the relevant driving pressure ratios of the order of 1.5, and the increased driving gas temperature. Estimates of gains in augmentation ratio due to elevated driving gas temperatures and of additional improvements in performance with higher driving pressure ratios by means of compounding or by the use of feedback ducts would be of interest.

J. A. Barnes[3]

The author is to be congratulated for his clear presentation of the operation and potentialities of pressure-exchanger dividers and equalizers.

He mentions in his Introduction that the low-pressure flow direction is the only one that reverses in the change from operation from divider to equalizer. Could he say anything about the effect of this on:

1 The losses in this port, which in the rig can only be angled correctly for one mode of operation?

2 The power absorbed by the rotor?

Turning to the performance maps Figs. 5 and 10, it is seen that the peak values of isentropic product efficiency are quite high—it would be difficult to better these values with turbomachinery with rotors of similar size. However, these high values only occur at modest pressure ratios and, for example, the pressure equalizer would appear to be unable to compete with an ejector under the conditions of high-pressure ratio at which the latter is usually required to work. On the other hand, the pressure divider would seem to offer a convenient way of taking fuel gas for use at low pressure from a pressurized pipeline; in this case the user, in taking his requirements, performs the service of boosting the pressure of the rest of the gas!

As the gas and rotor peripheral velocities involved are low, it is evident that these machines can be inherently robust and insensitive to erosion.

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[2] Associate Professor of Mechanical Engineering, American University of Beirut, Beirut, Lebanon.

[3] Imperial College, London, England.

Author's Closure

The author wishes to thank the discussers for their comments.

Concerning Dr. Azoury's point regarding the relative merits of feedback ducts versus compounding, compared with compounding, the prime virtue of feedback ducts is their simplicity. As a means of modifying pressure-exchanger performance characteristics feedback ducts suffer from the disadvantage that they are only effective in certain circumstances. The writer would, therefore, choose feedback ducts as an alternative to compounding for specific cases only, a particular example being the application to equalizers suggested in [4].

If an equalizer were employed as a thrust augmenter in a duct burning, or after burning, turbofan engine the thrust augmentation ratio attainable at the suggested driving pressure ratio of 1.5 would be approximately 1.9 with flow at a temperature of 3600 deg R supplied to the equalizer high pressure ports. The augmentation ratio would be reduced to approximately 1.5 by lowering the temperature of the high pressure flow to 2000 deg R. Internal cooling of the equalizer would be by the entrained low pressure flow which was assumed to enter the equalizer at 540 deg R.

In reply to Mr. Barnes, none of the ports at the LP end of the pressure-exchanger were angled in order to avoid complicating the variable geometry rig and also because of the flow reversal referred to by Mr. Barnes. It was, therefore, assumed that all whirl was given to incoming flows by the rotor, some of the power supplied by the driving motor being used for this purpose. A term equal to the whirl component of pressure was subtracted from the measured pressure of each of the outgoing flows in order to compensate for this work input. This correction was small due to the low rotor speed. Despite the correction, the effects resulting from poor flow in the region of the ports remained and it should be possible to obtain slightly improved performance characteristics with correctly angled ports.

The suggestion that the divider could be used as a device for alleviating pressure losses in natural gas distribution mains appears to be an excellent one. Two ways in which a divider could be incorporated are presented in Fig. 16. In both diagrams gas flows from a supplier A to consumers B and C. Consumer C draws gas, via a governor V_1, tapped from the main at a point between A and B. In both schemes some of the pressure drop which, without a pressure booster, would occur across the gover-

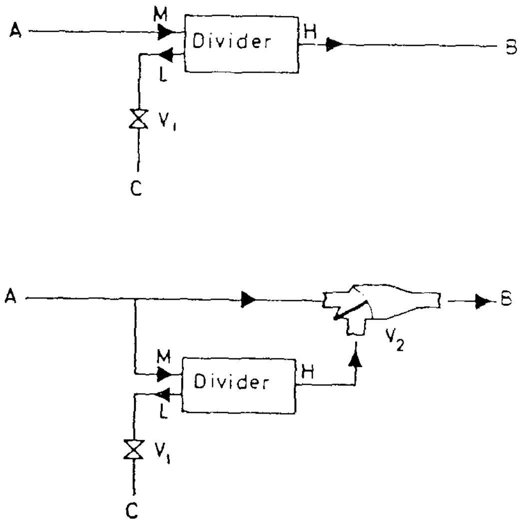

Fig. 16 Divider installation in gas mains

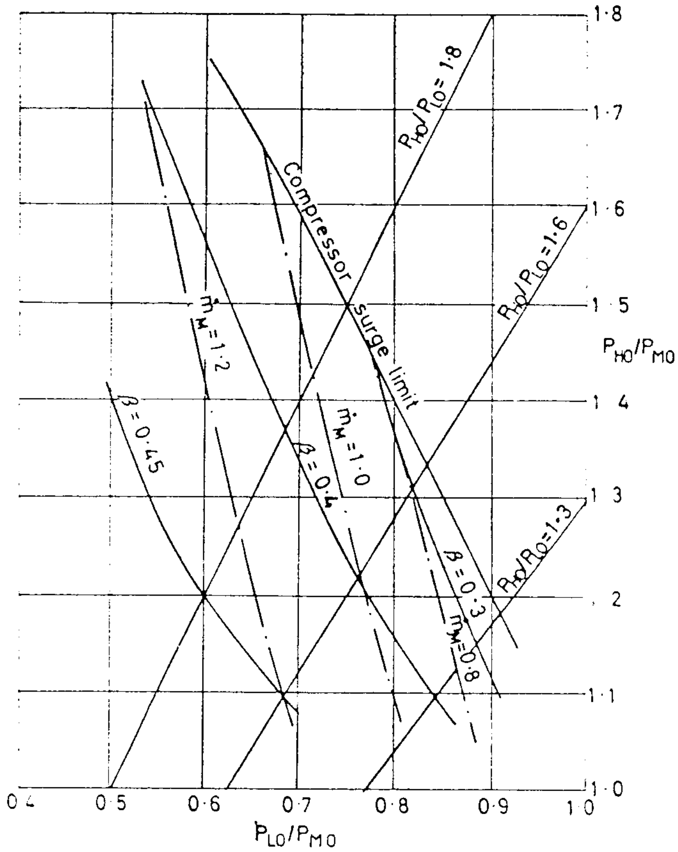

Fig. 17 Synthesized performance characteristics of a turbomachine divider, γ = 1.4

nor V_1 takes place in the divider with the beneficial result that the pressure of the supply to B is boosted thereby compensating for pressure losses in the main. Consumers B and C may be individual premises, commercial undertakings, or communities.

The arrangement shown in the upper portion of Fig. 16 should prove adequate for some applications, for example when C always takes a large proportion of the flow. The lower diagram illustrates a more flexible installation able to cope with a very wide range of demand by C. The double seating automatic valve, V_2, serves to ensure correct operation of the divider at start up and also that the full benefits of the divider will be realized when it is operating. The rotor of the divider could be driven by an electric motor, possibly controlled by a pressure switch connected to output C, or it could be made self-driving by appropriate angling of the inlet port.

The gas main pressure booster application is not one that appears to be suitable for a turbomachine counterpart of the pressure-exchanger divider. Fig. 17 shows a performance map synthesized from the performance characteristics of a centrifugal compressor and an axial flow turbine, both of fixed geometry, arranged as a divider. This configuration is comparable to the pressure-exchanger divider in complexity. A comparison of Fig. 17 with Fig. 4, a fixed geometry pressure-exchanger divider performance map, shows that the pressure-exchanger has greater operational flexibility than its turbo-machine counterpart. The lines of constant $\dot{m}_M$ in Fig. 17 should be compared with the constant M_M contours of Fig. 4. The mass flow $\dot{m}_M$ has been normalized by reference to the design point mass flow.

The use of natural gas instead of air as the working fluid is not expected to modify greatly the performance characteristics of the divider.

Shock Tube and Shock Wave Research / Proceedings of the 11th International Symposium on Shock Tubes and Waves, ed. Boye Ahlborn, Abraham Hertzberg, David Russell, Seattle, July 11-14, 1977 University of Washington Press, Seattle, 1978 p. 36-55.

THE PRINCIPLE OF THE PRESSURE-WAVE MACHINE AS USED FOR CHARGING DIESEL ENGINES

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Department Comprex Research

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ABSTRACT

The paper describes how energy from hot exhaust gases is transferred to air at atmospheric conditions by means of pressure waves.
It shows how pressure waves are produced in the machine and how these waves, produced as a continuous process, compress air and force both gas and air through the pressure-wave machine.
Special reference is made to the function of the pressure-wave machine at operating conditions differing appreciably from the design point.
A number of examples are quoted, as calculated with the aid of a computer program, and their results illustrated by wave and state diagrams.
Brief mention is made of the theory of non steady-state flow and the methods of calculation employed.

Symbols used

u = velocity of flow
p = pressure
ρ = density
T = temperature
a = speed of sound
s = entropy
R = gas constant
$\varkappa$ = ratio of specific heats
$\dot{V}$ = volume per second
x = distance coordinate
t = time

Indexes used

o = reference condition (atmospheric)
os = state on isentrope s at the pressure of the reference state o
s = isentropic change of state
G = exhaust gas
L = air
HP = high pressure
LP = low pressure

1. Introduction

The pressure-wave machine known as COMPREX® developed by Brown Boveri & Co., Baden, Switzerland for charging vehicle engines provides an alternative to the conventional exhaust-gas turbocharger. Its mode of operation, however, is basically different from that of the turbocharger. The exhaust gas and air are in direct contact with one another and the energy is transferred direct from one medium to the other.
Fig.1 shows schematically the combination of a Comprex and an engine.

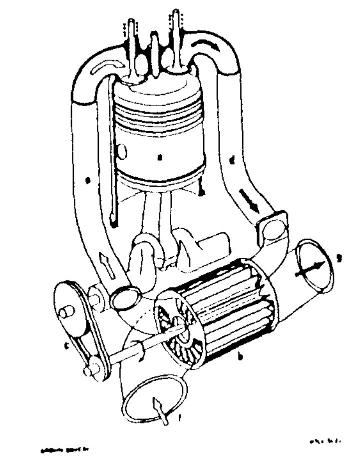


Fig.1: COMPREX pressure-wave machine, used for charging diesel engines

a = Engine
b = Cell-wheel
c = Belt drive
d = High-pressure exhaust gas
e = High-pressure air
f = Low-pressure air intake
g = Low-pressure gas outlet

The rotor is driven via a belt by the engine. For this only about 1% of the engine output is required to overcome bearing friction and windage losses. The rotor rotates between two casings with intake and outlet ducts for the engine exhaust and air. At the circumference of the rotor there are axial openings, open at the ends; they are referred to as cells.
Hot gas comes from the engine and flows through the high-pressure gas duct of the gas casing into the rotor. Here the energy is transferred to the air in the cell. The air is compressed, pushed towards the opposite end and forced through the high-pressure air duct in the air casing to the engine.
Before the exhaust gas has penetrated the rotor to such extend that it would flow out with the compressed air, the rotor cells are brought into connection with the outlet duct in the gas casing by the rotation of the rotor. Here roughly the ambient pressure prevails. The exhaust gas flows out of the rotor, expands, and the suction produced by the powerful flow is able to draw in fresh air through the low-pressure duct of the air casing into the rotor cells.
A new cycle begins.
Between the rotor and the Comprex the pressure pulsations in the piping resulting from the intake and outlet operations of the engine are damped to such an extent that the Comprex hardly notices them.
In the openings for incoming and outgoing flow before and after the Comprex the pressure is thus constant. Due to the rotary motion of the rotor, the ends of the cells are brought into contact with these openings in rapid succession in the course of a revolution. As a result, changes of state are caused in the cells, which are propagated right through the cells as pressure waves with the speed of sound.
These pressure waves, which are first produced in the rotor and have nothing in common with the inlet and outlet pressure pulsation of the engine, are responsible for the transfer of energy from the exhaust gas to the charge air. At the same time they accelerate and decelerate the contents of the cell and force the gas and air through the rotor: the gas to the exhaust pipe, the charge air to the engine. It is obvious that the peripheral speed of the rotor is an important factor, since its magnitude determines where a pressure wave produced at one end of a cell reaches the other end and hence also where in the rotor (relative to the casing openings) the air and gas are accelerated and decelerated.
Therefore it is by no means a matter of course that the gases which have entered the rotor are able to disappear through the exhaust again and it is consequently not obvious that the pressure-wave machine should be particularly suitable for charging those engines whose useful speed range is wide.
By suitably arranging the inlet and outlet openings along the periphery and as a result of the well-considered utilization of recesses in the end-face of the stators, opposite the rotor, it has proved possible to extend the speed range as well as the load range of the pressure-wave machine to such an extent that it is particularly well suited for charging such engines[1]. The functions of these recesses, or "pockets" as they are called, will be described in detail below.

2. Basic equations of non steady-state gas dynamics

If a small disturbance is caused in a pipe of constant cross-section, it will be propagated through the pipe with a velocity of dx/dt.
Before the disturbance the gas flows with the velocity u and has the state p, T, ϱ, s. Due to the disturbance the velocity is changed by du and the state by dp, dT, dϱ and ds (see Fig. 2).

If only frictionless flow without energy exchange due to heat transfer and conduction is considered, the changes in state are isentropic and, utilizing the gas equation $p = \varrho R T$ with the factor $a = \sqrt{(dp/d\varrho)_s}$ we obtain the following relationship between the state parameters:

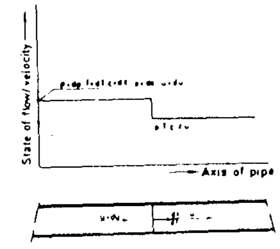


Fig.2: Pressure wave in a pipe

$$\frac{p}{p_0} = \left(\frac{a}{a_0}\right)^{\frac{2\varkappa}{\varkappa-1}} = \left(\frac{T}{T_0}\right)^{\frac{\varkappa}{\varkappa-1}} = \left(\frac{\varrho}{\varrho_0}\right)^{\varkappa} \qquad (1)$$

The factor a thereby introduced has the dimension of a velocity and can be used as a state variable. It is the velocity with which a deformation is propagated in an elastic medium, the speed of sound.

Applied to the disturbance in Fig.2, we can derive from the laws of conservation of energy, momentum and mass, with the aid of eq.(1), two important equations for non steady-state flow[2]:

$$\frac{dx}{dt} = u \pm a \qquad (2)$$

$$du = \pm \frac{2}{\varkappa-1}\, da \qquad (3)$$

Eq.(2) states that disturbances in a gas are propagated with the sum of the speed of flow and the speed of sound in the undisturbed gas.
From eq.(3) it follows that there is a linear relationship between the speed of flow and the speed of sound, of the form:

$$u \pm \frac{2}{\varkappa-1}\, a = u_0 \pm \frac{2}{\varkappa-1}\, a_0 \qquad (4)$$

The positive sign applies to pressure waves propagated to the right, the negative sign to waves propagated to the left.

Eq. (4) is also called the compatibility equation. Disturbances of large amplitude can be considered as consisting of an infinitely large number of small disturbances, each of which overruns its predecessor with the prevailing speed of sound.

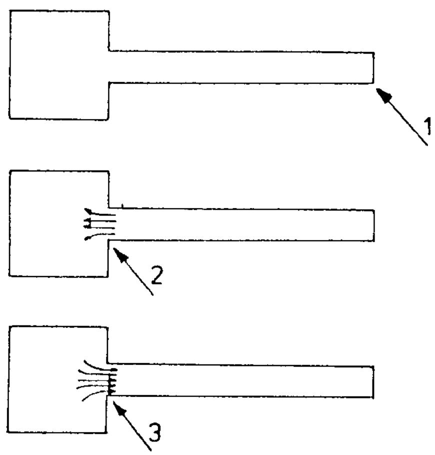
With the aid of eq.(2) and (3) it is possible to construct the propagation lines of the disturbances (also known as Mach lines) on a distance-time plane, this being known as the "wave diagram". This construction can be carried out in steps, enabling the changes in state and flow da and du to

be determined in a u-a plane (known as the state diagram) along straight lines, known as characteristics, with the directional coefficients $\pm \frac{2}{(\kappa - 1)}$

This method of illustrating the non-steady-state flow is generally called the method of characteristics[3].

2.1 Pressure waves in a pipe

In order to investigate pressure-wave phenomena in a pipe with the aid of the method of characteristics, it is necessary to know what conditions apply to the flow at the pipe ends. These so called boundary conditions are the relationships between state and velocity imposed on the flow from outside.
Here three boundary conditions are of particular importance:



1. End of pipe closed. No flow is possible. Illustrated in the u-a plane by the ordinate u = 0. (seetig 3)
2. End of pipe open. Outgoing flow into a space with constant pressure. The static pressure at the end cross-section of the pipe at all rates of outgoing flow will assume the level of the pressure in the space.
   Represented in the u-a plane by the straight line a = constant
   $(a = c\, p^{\frac{\kappa-1}{2\kappa}})$
3. End of pipe open. Incoming flow from a space with constant pressure. If this incoming flow is free from losses, we may write

   $$u^2 + \frac{2}{(\kappa-1)} \cdot a^2 = \text{constant} \qquad \text{(energy equation)}$$

   for the flow in the pipe, illustrated in the u-a plane by an ellipse whose tip coincides with the ordinate at the total pressure in the space.

These boundary conditions aplly to steady-state flow. Since the non steady-state changes in state are considered as quasi steady-state, the boundary conditions of steady-state flow can, to a first approximation, be used in calculating the non steady-state flow.
In a pipe, initially considered to be closed at both ends, we may now consider a disturbance caused by suddenly opening one end towards a space with a higher pressure. Pressure waves and changes in state can be traced with reference to Fig. 3.

Immediately after the pipe has been connected to the space with higher pressure, a pressure wave enters the pipe with the speed of sound in the static medium in the pipe (state 0).
This pressure wave causes a change in state. The pressure wave and boundary condition determine the new state. According to eq. (4), this new state is situated on the positively sloped straight line through 0 and the point

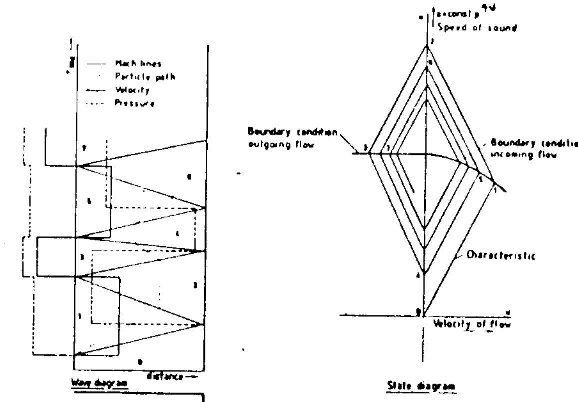


Fig. 3: Filling a pipe

of intersection with the curve of boundary condition for the incoming flow (state 1).
At the end of the pipe the pressure wave is reflected by the end surface and returns. In the state diagram this new state is found on the characteristic, now with a negative gradient, at the point of intersection with the boundary condition u = 0 (state 2).
On arriving at the open end of the pipe, the pressure wave meets a state of lower pressure (since $p_2 > p_1$). Expansion follows. The state 3 is again found with the help of a characteristic of positive gradient and the boundary condition, which stipulates that the pressure in the pipe shall diminish until it reaches the outside pressure. Gas flows out.
The change in state 2 - 3 is propagated from left to right as an expansion wave and accelerates the entire contents of the pipe to the speed of outgoing flow 3.
Arriving at the closed end of the pipe, the speed has to be reduced to zero, which is only possible when a new expansion wave is created, which reduces the pressure still further (state 4).
In the figure the path of a particle is constructed. This clearly shows that the speed of the particle and that of the pressure wave are very different. An incoming particle does not even reach the centre of the pipe when the pressure wave which caused its motion is already being reflected at the end. The pressure-wave process only comes to a standstill when the pipe is filled with the external pressure (5,6,7 ...).

By periodically changing the boundary conditions of the pipe it is possible to produce periodic pressure waves which can be utilized in a continuous process. The arrangement employed for this may be a pressure-wave machine.

Instead of having one pipe, it has several pipes (or cells) side by side round the circumference of a cylinder. The rotor thereby formed rotates between fixed casings, in which openings and faces alternately open and close the ends of the cell. The openings are connected with gases in different states. By rotating the rotor a periodical pressure-wave process is produced in each cell, the effect of which is that the media are changed from one state to another and can continuously flow in and out through the openings.

Depending on the field of application, there are various kinds of pressure-wave machine[4,5,6]. The Comprex machine described below is a pressure-wave machine employed for charging vehicle diesel engines.

2.2 Mode of operation of the Comprex

Fig. 4 shows the wave diagram and the corresponding state diagram for the pressure-wave process of a simple Comprex operating at full load and a medium engine speed. The rotor and casing are shown developed, so that rotation becomes a translation.

In the state diagram the curves of the boundary conditions for the four

openings in the casings are shown according to the conditions prevailing there. One cell is considered during the course of one revolution of the rotor.

Supposing the cells are infinitely narrow, so that they can be opened and closed instantaneously, and the changes in state are isentropic, also that reflections at the face between hot gas and cold air can be ignored and that the ratio of the specific heats $\varkappa$ is constant, the idealized pressure-wave process of this machine can easily be calculated using the method of characteristics, or graphically constructed[3], being fully described by the points 0,1 ... 7,8.
The representation of the pressure-wave process as shown in Fig.4 explaining the mode of operation of the machine agrees sufficiently closely to the actual process that takes place in the Comprex.

As soon as a cell containing fresh air is brough into contact with the high-pressure exhaust-gas opening, due to the rotation of the rotor, the engine exhaust gas, which is at a higher pressure, flows into the cell. It thereby compresses the air and the steady state in the cell is disturbed. The increase in the air pressure caused by the gas is propagated as pressure wave through the cell at the speed of sound. The air in the cell is compressed by the pressure wave and every particle of this air is accelerated. The wave diagram in Fig. 4 shows the path of this pressure wave, while in the diagram of state the changes in state caused by the pressure wave are found at the point of intersection of the characteristic with positive gradient from 0 with the curve of the boundary condition for the incoming flow of exhaust gas into the cell (state 1).

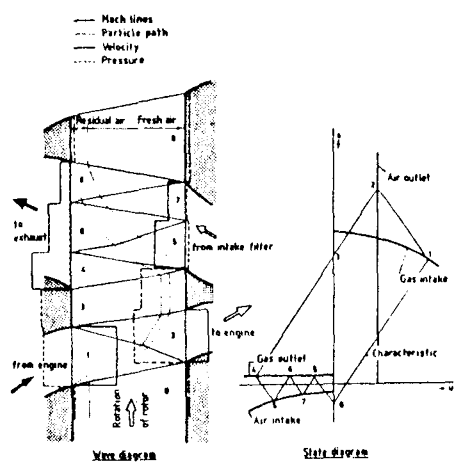


Fig.4: Pressure-wave machine without pockets
Full-load design speed (without losses)

When the pressure wave has passed through the cell, the pressure and speed prevailing throughout the cell are those of state 1. The gas and compressed air now flow towards the air end, but much slower than the rate at which the pressure wave is propagated through the air. The path of the first incoming gas particle (Fig. 4a) represents the front separating the hot gas and cold air. The pressure wave reaches the air end of the cell in the very moment the cell, moving with the rotor, comes opposite to the high-pressure air duct.
The compressed air is now able to flow out. The opening edge for the high-pressure air duct, seen in the direction of rotation of the rotor, must be displaced with respect to the opening edge of the h.p. gas duct by as many degrees as the cell under consideration can cover in the time the pressure wave 0-1 covers it.

This important "matching condition" is augmented by some more, details of the significance of which for the operation of the Comprex being dealt with later.

Hence the compressed air in the cell has the same velocity and pressure as the incoming hot gas (state 1). But the density of the air is greater than that of the gas. If we assume that the h.p. openings at the air and gas ends are of equal size, a cell passing the openings can accept gas and deliver air, during equal intervals of time. Consequently more air would flow out of the Comprex per unit time than the amount of incoming gas.

This is impossible since, apart from a small amount of additional fuel, the amount of exhaust gas coming from the engine is equal to the amount of charge air fed to the engine. Hence it appears, the volume flow of air is less than that of the gas. In other words, the speed of outgoing flow of the charge air leaving the cell is less than the speed of exhaust gas entering the cell. The speed of outgoing flow is determined by the volume drawn in by the engine per second and thus represents a boundary condition for the outgoing flow from the cell; it is less than the speed of state 1. According to the laws of non steady-state gas dynamics (eq. 3), this reduction in speed imposed from outside, results in a rise in pressure inside the cell.
In the state diagram in Fig. 4 the new state 4 is on a characteristic with negative gradient at the point of intersection with the boundary condition for flow out of the cell. The resultant positive pressure difference between charge air and exhaust gas is clearly expressed in the diagram of state. For the engine a large pressure difference is favourable as regards fuel consumption because the work of gas exchange performed by the piston is directly dependent on the back-pressure of the exhaust gas. In engines with valve overlap the positive pressure difference is effective during scavenging. The pressure increase now begins to move back as second pressure wave through the oncoming contents of the cell until it reaches the gas end. Each particle of air and gas in the cell is now retarded by it to the speed of state 2 and compressed to pressure 2. At the same time compression wave 1-2 is carried by the rotor, which continues rotating, and reaching the gas end the cell has just passed the h.p. inlet opening. Now the cell is closed. The size of the inlet opening is thus determined by the peripheral velocity (i.e. the speed) of the rotor and the transit time of the pressure wave which, according to eq. (2), is dependent on the speed of sound and the speed of flow: a second "matching condition". Now, suddenly, nothing more is able to enter the end cross-section of the cell. State 2 in the cell is disturbed by the left-hand casing wall, which blocks the flow. The new state 3 is situated on the characteristic with positive gradient from 2 at the point of intersection with the boundary condition $u = 0$. The exhaust gas can expand.
This expansion is propagated through the cell with the effect that the entire contents of the cell expands and comes to a standstill: state 3. The expansion wave 2-3 arrives at the air end in the very moment the end of the cell finishes crossing the air outlet opening and the cell is closed. This determines the size of the outlet opening: a third "matching condition". Now the casing wall after the outlet opening does not cause any further change in state because the condition $u = 0$ is already fulfilled by the arriving expansion wave.
The cell is now closed at both ends, the pressure in it has dropped below the pressure of the charge air, though it is still considerably higher than the ambient pressure; the contents of the cell are at rest. In the wave diagram in Fig. 4 the path of the first incoming gas particle is shown. It indicates to what extent the exhaust gas enters the rotor. It can also be seen that not all the air originally in the cell flows out through the outlet duct. Some air remains in the cell, being later used for scavenging the rotor. In the gas casing the outlet opening is positioned after the inlet opening. It is connected to the exhaust pipe.
As soon as the end of the cell at the gas end is open to this low-pressure output duct, exhaust gas flows out of the cell. It expands to the pressure in the exhaust pipe, which is always slightly higher than atmospheric. In the state diagram this expansion takes place along the characteristic from 3 which intersects the boundary condition for outgoing flow at 4.
The drop in pressure is propagated through the cell again as expansion wave 3-4 towards the air end, the entire contents of the cell assuming the pressure and flow velocity of the outgoing gas, state 4.
At the air end the opening edge of the l.p. air duct, seen in the direction of rotation of the rotor, must be displaced with respect to the opening edge of the l.p. gas duct, to such an extent that the cell is just beginning to open in the very moment the expansion wave arrives: a fourth "matching condition".
Owing to the pressure drop through the intake filter, the pressure in the air inlet duct is less then ambient and the contents of the cell will ad-

just to this pressure.
The characteristic 4 and the boundary condition for incoming flow of fresh air determine the state 5 at the air end of the cell. A new expansion wave 4-5 is produced. It travels to the gas end, expands gas and air still further and reduces their velocity. This velocity (5) with which the air now flows into the cell is still high enough to enable the cell to be re-filled with fresh air. In the wave diagram the path of the first incoming air particle is shown. When the expansion wave arrives at the gas end, state 5 is prevailing throughout the entire cell. Outside the cell in the exhaust pipe the pressure is slightly higher. A characteristic 5-6 with positive gradient in the state diagram shows how the pressure and speed in the cell vary under the influence of the boundary condition "gas outlet". This change travels as compression wave 5-6 back through the cell to the air end. The whole contents of the cell are retarded, as shown in the wave diagram by the shape of the particle paths, and compressed to state 6. Arriving at the air end, the compression wave brings state 6 in connection with the air inlet duct with the cell fully opened. In this duct the pre-vailing pressure is lower, a change in state from 6 to 7 takes place: drop in pressure, drop in speed. The speed of the incoming flow is reduced. An expansion wave travels back and, as before, changes into a compression wave at the gas end, giving rise to state 8. At this instant all the ex-haust gas has left the cell. In state 8 some of the air flows out, that did not reach the engine after compression by waves 0-1 and 1-2. The rest will not flow out until the next cycle. This air flowing into the exhaust pipe is known as scavenging air, with which the rotor is cooled. The last compression wave 7-8 reaches the air end at the very moment the cell is just closed: Fifth "matching condition".
No more air is able to enter. The boundary condition "cell closed" compels a change in state to take place as prescribed by the characteristic 8-0. A final expansion wave is produced. The contents of the cell are expanded to the initial pressure, the velocity becomes zero. The cell is closed at the gas end at the very moment this state is reached in the entire cell: Sixth "matching condition". A new cycle can now begin.

The process described in the foregoing can only function when the "matching conditions" are fulfilled. This can only take place at one working point: at a definite speed (peripheral speed of the rotor), at a definite through-put (rate of flow) and a definite load (temperature of exhaust gas corres-ponding to the speed of sound). This is because a change in only one of these quantities alters the transit time of the pressure waves, thus causing a mismatch in the process. In this connection it is usual to refer to the "design point" and "matched speed".

By employing the pockets in the stator walls, referred to in part 1, the pressure-wave machine can be rendered less sensitive to mismatch, thus enabling the ranges of load, speed and volume to be appreciably enlarged. How these pockets function can be explained, in much the same way as for a matched engine, by the wave and state diagrams for different working points of the pressure-wave machine.
However, the construction of such diagrams is in most cases rather diffi-cult and takes considerable time because each mismatch of an edge in the process gives rise to a new pressure wave, which has to be carried right through the process. The work involved can be greatly reduced with the aid of a computer program.

3. Method of calculation, computer program

In order to calculate a realistic pressure-wave process, for instance for a Comprex machine with pockets, for unmatched speed, it must be possible to take into account the effect of friction, the mixing of gas and air, throttling actions when the cell is opened and closed, losses due to flowing past a narrow wall between two openings, surges, reflections at the separating front between hot gas and cold air, as well as the influence of centrifugal force on the mass distribution in the cell.
The program developed by BBC, with which variations in the flow and the states in a cell can be calculated, employs the method of differences put forward by Lax and Wendroff[7].

The great advantage of this method from the programming point of view is that solutions do not have to be sought along Mach lines and particle paths - the shape of which form part of the problem itself - but are deter-mined at the points of a fixed grid lattice.
The flow in the pipe is regarded as being one-dimensional. This implies that the state parameters are constant across a plane at right-angles to the axis of flow. In the basic equation of one-dimensional flow (differential equation for the laws of conservation of mass, momentum and energy) it is possible to take into account deviations from this ideal flow pattern by suitably adapting the disturbance terms in the differential equations[8].

The pipe (in our case the cell) is uniformly divided into a large number of lattice points. The change of state between the points is assumed to be linear.

If, at the start of the process, the states at all points are known, the speed, pressure, temperature and speed of sound in the medium can be calculated by this method for all points in the lattice, except the boundary points for each time step of the calculation.
The states at the boundary are determined by the method of characteristics, by relating the boundary conditions with the compatibility equations (2) along the Mach-Lines running towards the boundary points.
Estimation of the various coefficients included in the program to allow for the above-mentioned influences and which enter the calculation as inputs, remains subject to theoretical and experimental research[9].
Efforts are being made to utilize the program such that it is possible to distinguish between the influencing quantities and their effect with ever increasing accuarcy.

4. Numerical examples of the effect of pockets

For a number of typical Comprex working points the non steady-state variation of the state parameters in the rotor were calculated with the computer program.

In order that the results of the calculation and the associated explanation of the pressure-wave process may be illustrated as simply as possible, the geometry chosen in these examples for the openings and pockets differs slightly from the optimized geometry of control edges as used for certain service applications by BBC. Therefore the calculated results can only be regarded as possessing qualitative value.

From the results it is evident that on widening the operating range of a Comprex without pockets, based on the "design point" (matched speed, full load) it is by no means a matter of course that the machine will function properly, because scavenging is no longer assured. This implies the dis-placement of the expanded exhaust gas from the cell by the incoming fresh air. When pockets are introduced, the Comprex is then adequately scavenged and can be operated throughout the entire no-load range of the engine as well as at full load, producing a high pressure ratio, through the full, wide speed range.

The diagrams plotted by the computer show the speeds and pressure curves at both ends of the cell in the diagram as a unrolling of rotor and casings (Fig. 5a).
In order to assess the computed results qualitatively and compare them with other examples, we resort to the graphical method of characteristics. The aim is to represent the changes in state caused by pressure waves in a corresponding state diagram. Since the pressure-wave process is computed, taking into account the major losses that occur, the changes in state are no longer isentropic. Therefore plotting the changes in state in a u-a plane will be different from the isentropic case of Fig. 4 (chapter 2.2).
In order to take into account differences in entropy in the cell, as are already given by the presence of the media gas and air, the state diagram is plotted in non-dimensional form $\frac{a}{a_{0s}}$, $\frac{u}{a_0}$.

The compatibility condition (4) can be written for isentropic changes in state in gases with different entropies as follows:

$$\frac{u}{a_0} + \underbrace{\frac{2}{\kappa - 1} \cdot \frac{a_{os}}{a_0}}_{\text{gradient}} \frac{a}{a_{os}} = \text{constant} \qquad (5)$$

The characteristics in the $\frac{a}{a_{os}}, \frac{u}{a_0}$ - plane can consequently have different gradient, depending on the entropies of the gases. In the a-s diagram the changes in state take place along different isentropes.

In the non-isentropic pressure-wave phenomena, taking losses into account, as in these examples, the compatibility conditions (5) contain additional terms for friction and heat transfer[2] and are therefore no longer linear.

The lines of progression of the disturbances (pressure waves) are no longer straight lines because the temperature (and consequently the speed of sound varies aprreciably in the cell.
In the wave diagrams now the curves at the ends of the rotor are divided into areas of more or less constant state, which are shown in the diagrams of state as points on the corresponding boundary condition curves. The lines in the wave diagram form an optical bridge between the change in state at one end of the rotor (cause) and the change in state at the other end (result), thereby depicting the effect of pressure waves on the process.
The lines in the diagram of state, which can only be regarded as compatibility conditions to a very rough approximation, indicate the rule according to which the states must change.

4.1 Full load, design speed

Fig. 5 shows the results of the calculation for a Comprex process at full load (exhaust gas temperature over 600°C) and the design speed. Given is the volume of air drawn in by the engine, the temperature of the exhaust gas (corresponding to the load point of the engine) and the speed of the Comprex.

In the diagram of state (Fig. 5b) the regions of constant state are taken from the wave diagram as points, 0, 1, ... 8.
Despite the considerable difference between the method of characteristics

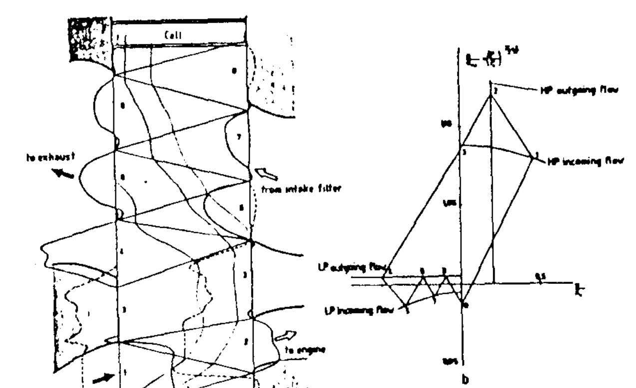


Fig 5: Full-load, design speed (without pockets)

for the ideal case of flow free from losses and the Lax-Wendroff method for the realistic case of flow affected by losses, a comparison with Fig.4 shows that the way of illustration the process by means of simple straight lines showing the changes in state and the influences of pressure waves may remain unchanged.

4.2 Full load, reduced speed

Fig. 6 shows the results of calculation for a full-load point of a Comprex without pockets at 50% of the matched speed. It was assumed that the cell, before being opened to the h.p. gas duct was filled with fresh air at atmospheric pressure (as if operation with efficient scavenging were possible).

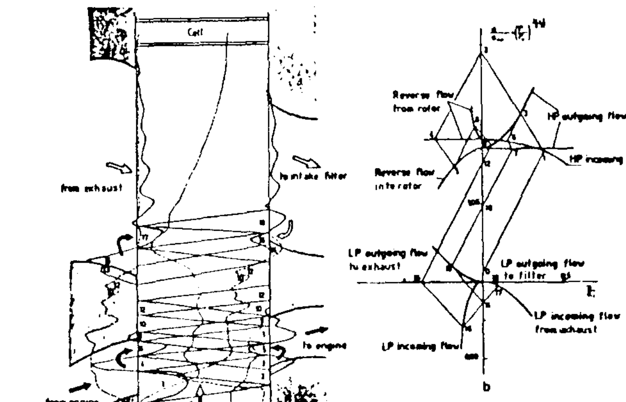


Fig 6: Full load, low speed, without pockets

The pressure-wave process is now no longer matched. The first pressure wave, which is produced in the cell when it is opened to the gas duct reaches the air end before the cell end reaches the air outlet opening, owing to the low speed of the rotor.

State 1 in the cell, resulting from state 0 when the exhaust gas flows in at the left-hand end of the cell, is changed to state 2 when the pressure wave 0-1 is reflected at the wall. In the state diagram in Fig. 6 this is depicted by a straight line with negative gradient from 1 to the boundary condition u = 0.
The air is compressed.

The compression wave travels back to the gas end, arriving there at an instant when the cell is still open to the gas inlet duct. The high pressure 2 will decrease to the pressure in the exhaust duct; gas will flow out (state 4). This reverse flow of gas causes charge air to flow in at the air end (state 5). The pressure and velocity of state 5 are given by the compatibility condition between states 4 and 5 and the boundary condition for the air outlet opening. The stipulation of this boundary condition for an unmatched pressure-wave process is not so simple as for a matched process, as will be explained below.

Here, too, the volume of air drawn in by the engine is a condition for the mean outlet speed of the charge air from the cells.
Owing to the action of the pressure waves caused by mismatch, areas of different size of different states are produced in the outlet opening (see Fig. 6). Now the pressure established in the air outlet duct will be such that the mean value of velocity in 3, 5, 7 and 9 provides the volume of air required by the engine. At the same time the boundary condition for incoming flow of gas will be established at that pressure at which the mean velocity in 1, 4, 6 and 8 is greater than the mean charge-air outlet velocity in the ratio of the densities of the gas and the air.

Consequently the charging pressure level that is established is relatively high. At the end of the cycle, however, it is found that there is no scavenging at all. This in contrast to what was assumed. The rotor becomes filled with exhaust gas.

The relatively high pressure in the cells following the high-pressure

section (state 12), indeed allows the gas to flow out into the gas outlet duct when opened (13, 15), but the expansion wave thereby produced (change in state from 13 to 14) is not carried far enough by the slowly moving cell to be able to draw in fresh air from outside on arrival at the air end. The wave strikes the wall where it creates a pressure drop (14). This is propagated towards the gas end where it draws exhaust gas back into the cell 17.

In the succeeding cycle, instead of compressed air, compressed gas flows into the engine, which then stalls. Operation with the Comprex is not possible.
The drop in pressure at 14 in Fig. 6 could be avoided if, at that wall section in the machine, an incoming air flow at sufficient pressure into the cell could be produced.
This desired effect can be provided by means of a recess in the wall, known as an expansion pocket. This pocket can accept air from the cells in the areas 10, 12 (see wave diagram), since here the pressure level is relatively high. Now this air can flow back into the cells from the area 14.

4.2.1 Expansion pocket

The result of calculation of a cycle using such a pocket is shown in Fig. 7.
In the diagram of state (Fig. 7b) it can be seen that the pressure resulting from the outgoing flows 7 and 9 at the opposite wall (area 10) is high enough to enable air to flow out into the pocket (area 11). The expansion wave starting from here (10-11) reduces the pressure at the gas end of the cell. An initially weak outgoing flow of exhaust gas takes place when the cell is opened to the low-pressure outlet duct (12). A second expansion wave (11-12) causes air being drawn in from the pocket at 13, thus making possible a powerful outward flow of gas from the cell (14).

The boundary condition "expansion pocket" is situated in the diagram of state at the pressure level at which the sum of the incoming and outgoing flows (11 and 13) is zero. Areas 15, 16, 17 subsequently result in the rotor being scavenged and the cells filled with fresh air. The cycle becomes stable, the engine does not stall.

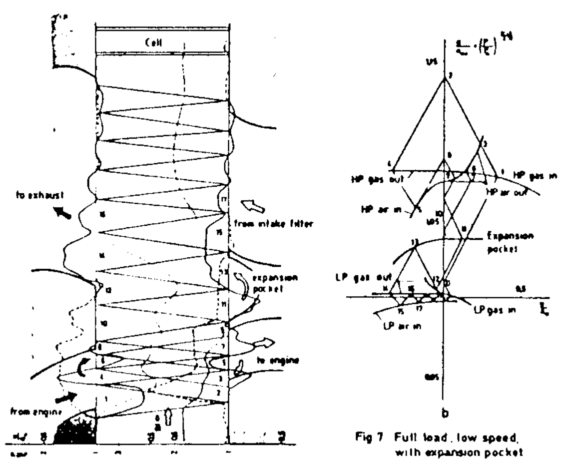


Fig 7 Full load, low speed, with expansion pocket

If we consider the pressure-wave phenomena for this Comprex with expansion pocket in the h.p. section, we notice the areas of reverse flow into both openings, which interrupt incoming and outgoing flow (for an explanation see reference to Fig. 6). It is obvious that such strong areas of reverse flow are bound to exert an adverse effect on the transfer of energy in the Comprex.

Reversing the direction of velocity merely causes additional [illegible] the flow.

4.2.2 Compression pocket

By means of a recess in the casing wall before the air outlet duct, known as a compression pocket, it is possible to eliminate these areas of reserve flow. The charging pressure attains a much higher level due to the higher efficiency of compression.
The cycle sequence can be easily followed with the aid of Fig. 8. In the pocket there is a pressure which is higher than that in the cells at the end of the scavenging phase of the preceding cycle. Air flows out of the pocket into the cells (state 1). Slight pre-compression takes place (0-1). The initial incoming flow of engine exhaust gas (state 2) now takes place - in contrast to the case without a compression pocket (Fig. 7) - at a low velocity of incoming flow. However, the pressure wave 1-2 starting here is able to escape into the pocket and is the real cause of the higher pressure, mentioned previously, in the compression pocket.

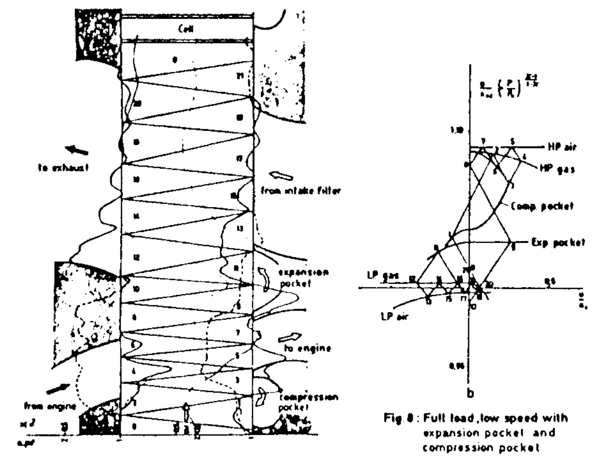


Fig 8: Full load, low speed with expansion pocket and compression pocket

The pressure will be established at a level at which the incoming and outgoing quantities 1 and 3 are equal.

The expansion 2-3 thereby produced permits a strong incoming flow of gas into the cells at 4, as well as a subsequent outgoing, instead of reverse flow at the air end, at 5.

The cycle thereby established, as can be seen in the diagram of state, creates a higher charging pressure level (see Fig. 6), which may be attributed to the lower losses in the high-pressure part.

4.3 Low load, matched speed

With a Comprex having no pockets, operation at low load (exhaust gas temperature less than 300^{0}C) is not possible, as may be seen from Fig. 9.
From the wave diagram (Fig. 9a) it is evident that the exhaust gas can only flow out through the charge-air duct, so that hardly any fresh air is able to enter the cell. In the diagram of state it will be seen that the low pressure at 8 and 10 is responsible for the poor scavenging effect (areas 13 and 16).

This low pressure is a consequence of the fact that the difference in density between exhaust gas and air in the cell, after compression by the first pressure wave 0-1, is [illegible] account of the low temperature of the exhaust gas.
Quite apart from the fact that mismatch in the cycle at the various opening and closing edges causes pressure-wave effects, which are propagated through

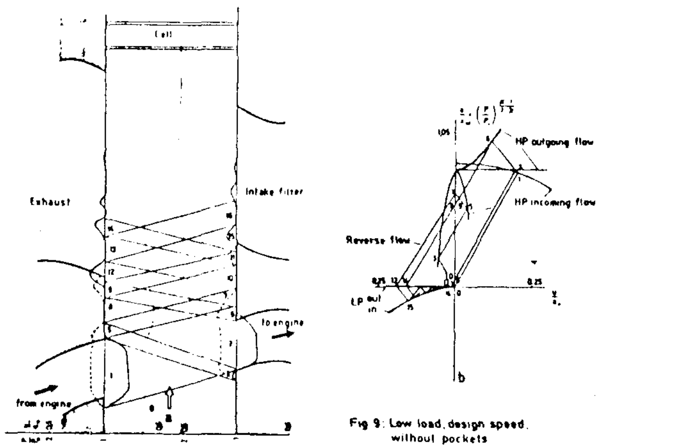


Fig 9: Low load, design speed, without pockets

the cycle, the transfer of energy between gas and air takes place according to the same pattern as at the full-load design speed. Slight deviations from the true "match" (area 3,6) are a consequence of the lower operating temperature and hence of the lower speed of the pressure waves.

Let us assume that the cells are full of fresh air before the cycle starts. Opening the cell to the exhaust gas duct causes a change in state 0-1 in the cell. This change in state is propagated through the air, which is compressed to state 1 and flows through the cell at the same speed with which the gas flows into the cell.
Arriving at the open air end of the cell, state 1 has to adapt itself to the boundary condition imposed by the engine. This condition is determined by the volume flow drawn in by the engine. Thus the speed of outgoing air flow from the rotor is pre-determined. The quantity of air that flows out is roughly equal to the quantity of gas flowing into the Comprex. At low load the density of the air is only slightly higher than that of the (only lukewarm) exhaust gas, so that the velocity of the charge air flowing out of the Comprex must be less than that of the incoming gas, in the ratio of the respective densities. But since the cross-section available for the outgoing flow - as can be seen from the wave diagram - is smaller, owing to the mismatch at the air outlet opening edge, the outgoing velocity of the air will diminish less than proportional to the density ratio.

The pressure of the new state 2 (on the characteristic with negative gradient from 1 in Fig. 9b) can therefore only be slightly higher than the pressure of state 1. When, following this, the change in state 1-2, which is again propagated through the cell, arrives at the gas end, the cell is closed. The change in state demanded by the boundary condition "wall" (velocity = 0) takes place along a characteristic whose gradient is now positive in the diagram of state. From this it is quite clear that the pressure at the wall has to be severely reduced when the speed is retarded.

If the pressure in the cell before opening to the low-pressure outlet duct is only slightly above the exhaust pressure level, it is not possible for sufficient expansion to take place when the cell is opened. Little gas flows out and at the other end too little air can be drawn in. The incoming gas cannot all flow out and after only a few revolutions of the rotor the cells are full of gas (see Fig. 9a). The difference in density between incoming gas and the gas collecting in the cells thus becomes even smaller.

We thus have the explanation for the fact that states 1 and 2 almost coincide with one another, as shown by the results of calculation after a number of cycles in Fig. 9b, and as a result the pressure in areas 8 and 10 of the wave diagram drop almost to atmospheric level. Nothing is able to flow into the gas outlet and air intake ducts.

4.3.1 Gas pocket

An effective counter-measure is to create a pocket in the wall between inlet and outlet openings in the gas casing (known as the gas pocket).
By feeding this pocket with exhaust gas from the engine, the pressure in this part of the rotor is increased and the prerequisite conditions are created for scavenging taking place.
This feed can be effect, for instance, by setting back the end face between the gas duct and the pocket, thus permitting flow from the exhaust gas duct into the pocket.

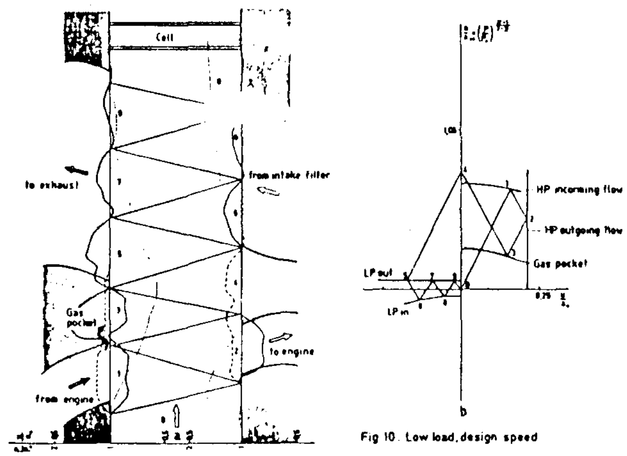


Fig 10. Low load, design speed

The calculated results of this example are to be seen in Fig. 10. Since now, in comparison to the case where there was no gas pocket, the gas coming from the engine can now flow into the rotor through a larger cross-section (existing duct plus gas pocket), the pressure of the gas coming from the engine is lower when it enters the Comprex. When the cell is opened to the gas intake duct, the velocity in the cell will also be smaller.

The corresponding state 1 is situated on a straight line with positive gradient through 0 at the point of intersection with the boundary condition curve given by the pressure level of the gas before the Comprex. The pressure wave 0-1 has compressed the air in the cell to the pressure of the incoming gas and accelerated to the latter's velocity. The difference in density between gas and air is small (low load, low gas temperature). If the rate of outgoing flow into the air duct were to remain unchanged, in the time during which a cell crosses the outlet opening, less air would reach the engine than the amount of gas coming from the engine at the other end, for because a larger cross-section is available for the gas entering via the gas pocket.

The state of the outgoing air therefore has to be changed when pressure wave 0-1 arrives at the air end. The new state 2 is situated on a straight line of negative gradient through 1 at the point of intersection with the boundary condition (Fig. 10b). This boundary condition is again the speed of the air leaving the rotor, determined by the geometry of the outlet duct and the volume of air drawn in by the engine per second. This speed is greater than that of the incoming exhaust gas (small difference in density with a large difference in cross-section). As can be seen in Fig. 10b, the pressure of state 2 is lower than that of state 1.

The pressure wave which transfers this change in state 1-2 through the cell to the opposite end (an expansion wave) meets the boundary condition "gas pocket". The pressure in the gas pocket will be lower than that in the gas inlet duct owing to throttling losses on entering the pocket. The speed at which the gas flows out of this pocket into the cell can be found from the diagram of state at the point of intersection between a charac-

teristic line with positive gradient trough 2 and the curve of the boundary condition.

This pocket causes the change in state 2-3 which is propagated as expansion wave to the air end bringing the contents of the cell to state 3 and, on arriving at the air end is converted into a compression wave 3-4 by the boundary condition "wall". This wave travels back and compresses the contents of the cell to state 4.

When the wave reaches the gas end, the gas is able to expand - because the cell is open to the gas outlet opening - and flow out at the speed of state 5 (on the line 4-5 and the boundary condition "exhaust back-pressure"). As a result the states 6,7,8,9 are created in the low-pressure part in the corresponding areas of the wave diagram in Fig. 10a.
The gas is able to flow out almost completely. Fresh air enters the cells and the combustion air drawn in by the engine during the next cycle will contain only a small amount of exhaust gas.
When fitted with a Comprex having a gas pocket, the engine can be operated through the entire no-load range with the help of the gas supplied by this pocket.

4.4 Full load, design speed with all pockets

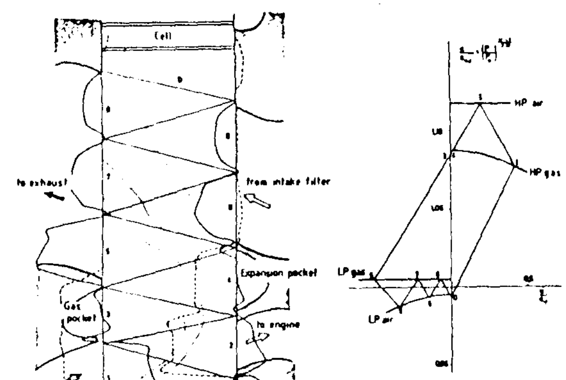


Fig 11 Full load design speed, with compression and expansion pockets and gas pocket

Finally, Fig. 11 shows the result of calculation for the design speed at full load when the Comprex is equipped with all the pockets described in the foregoing. It can be seen that the idea of pockets as a means of maintaining the pressure-wave process at unmatched speeds and low loads, is not a compromise detrimental to the optimal performance at the design speed. The same pressure levels as without pockets are reached. The wave diagrams and the diagrams of state for the two cases resemble one another closely (see Fig. 5).

5. Conclusions

From the calculated examples of different operating points it is evident that operation with the Comprex is by no means confined to full load and a narrow band on either side of a definite speed, as might be imagined from the description in part 2.2.

With the pockets it is possible to maintain the pressure-wave process at speeds differing appreciably from the value to which the Comprex is matched, and also at low engine loads.
On the other hand, at unmatched speeds the mismatch of the pressure waves at the various opening and closing edges on machines not equipped with pockets, results in a failure of scavenging.

From the examples it can be seen that each pocket is effective at a certain defined operating point. Altogether, they do not adversely affect one another. Neither do they affect the optimal operation of the Comprex at its design point.

Summarizing, the following general rules can be established regarding the tasks of the pockets:

- The expansion pocket is used to maintain scavenging at full load and low speeds
- The gas pocket enables the Comprex to operate at low loads and intensifies scavenging at all unmatched operating points at full load.
- The compression pocket improves the efficiency of compression at full load and low speed.

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WAVE ROTORS FOR TURBOMACHINERY

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ABSTRACT

Potential applications of wave rotors as turbomachinery components are reviewed. Investigations and development efforts which have been conducted over the past forty years have identified a number of areas for which the wave rotor appears to have unique operating characteristics relative to conventional turbomachinery. The potential impact of wave rotors in these applications, associated wave rotor configurations, and the status of wave rotor concepts that have been studied are evaluated and summarized.

INTRODUCTION

Wave rotors are high throughput, axial flow machines that perform the dual functions of compressors and turbines in a single rotor. These devices are based on direct contact between high and low pressure working fluids to transfer work between them via unsteady compression and expansion waves. They exhibit superior wall cooling capabilities, allowing them to operate at higher peak cycle temperatures compared to advanced gas turbines. When combined with good component efficiencies, wave rotors can perform a significant role in modern turbomachines as an efficient, high pressure topping stage to improve their overall performance.

Claude Seippel first introduced the notion of a wave rotor as a topping stage in 1949 for a locomotive gas turbine engine [1]. While somewhat crude and inefficient, this combined cycle device functioned and provided the first success for this type of application. Over the intervening years, a wave rotor concept called the Comprex™ has undergone considerable development by the Brown-Boveri Company [2]. Even though the Comprex is designed for supercharging Diesel engines, it performs basic functions similar to that desired in the topping cycle application: To increase the gas pressure before adding the heat of combustion, thereby increasing the amount of work that can be extracted from the cycle per unit mass flow of air.

In the locomotive topping cycle application, the wave rotor takes partially compressed air (from a conventional axial flow compressor, for example) and raises its pressure still further. This air is transferred to a combustor, which heats the gas to the peak cycle temperature. In one version of this cycle, the hot combustion gas flow is split, with the highest temperature part of it re-entering the wave rotor (called a pressure exchanger) to perform the work of compressing the intake air, and the other part of the combustion gas is sent at somewhat lower temperatures to drive a conventional gas turbine. The net effect is a higher pressure, higher temperature cycle which is thermodynamically more efficient. In a second version of this cycle, a different kind of wave rotor (called a wave rotor/turbine) processes all of the combustion gas internally, both to compress the incoming air and to extract shaft power.

A host of other applications of wave rotor technology have been considered since Seippel's first machines. These include the basic concept of the wave rotor as a topping cycle applied to stationary electric power plants [3,4], to marine gas turbine propulsion units, and to aircraft turbine engines [5]. In each of these cases, exploratory research and, in some cases, initial developmental research has been carried out. These efforts, described below, throw an interesting light on the unique attributes of this technology and form the basis for more recent developments that have led to its consideration for small gas turbine aircraft engines.

The unifying theme of the applications and development history reviewed here is the use of the wave rotor in which there is a competition between an additional component (with its finite component efficiency) and the potential increase in peak cycle temperatures and pressure which determines the final cycle efficiency. There are cases in which the benefits are considerable and others in which the addition of the extra wave rotor component is clearly a debit. A somewhat different, but equally interesting, perspective on gas turbine applications of wave machines is given in the 1973 review by Jenny and Bulaty [6]. Many of their conclusions are echoed here and strengthened by calculational developments and new test data obtained in the last ten years.

STATIONARY POWER SYSTEMS

Thayer and his co-workers have reviewed the utility applications of the wave rotor to gas turbine topping cycles, to MHD cycles, and to pressurized, fluidized bed (PFB) power systems [4]. These examples represent three distinct advanced power systems which utilize wave rotor technology. The first of these three appears to offer a significant advantage in cycle efficiency. All three applications increase the plant reliability (e.g., by reducing turbine wear) and show the potential for cost savings.

The use of the wave rotor as a high temperature expander was examined for stationary power plants by Weatherston and Hertzberg [3]. They observed that a nearly isentropic series of compression and expansion processes could be achieved for transmitting work from the higher pressure gas to the lower pressure gas when the two gases are impedance matched; that is, if their molecular weights are chosen in proportion to their respective temperatures. They also developed the basic system layout for the topping cycle applications involving the wave rotor as the high pressure stage, producing a closed loop, high pressure flow of steam to a power take-off turbine. Their work was recently updated by Zumdieck et al. [7] who investigated this same application to combined cycle/coal gasification plants. Subsequent calculations indicated that a substantial gain in overall cycle efficiency could be achieved when a wave rotor was used in an open cycle mode with the use of clean fuels. Also, the wave rotor and the accompanying power turbine both could use present-day metallurgy even though the peak cycle temperatures were well above the capabilities of current turbine cooling technology. These results

confirm the benefits of the wall cooling capabilities of the wave rotor to advance power systems to higher peak cycle temperatures with a minimum (or zero, in this case) of materials development.

In the MHD application, the wave rotor is used as a high temperature compressor. Normally, high cycle temperatures would be reached by regenerative heating and the use of oxygen-enriched air. By regeneratively heating a lower pressure gas and then compressing the heated gas to high pressure with the wave rotor, a savings in oxygen enrichment is achieved and the regenerative heat exchanger can operate at lower materials temperatures. In this instance, the wave rotor would be driven by high pressure steam from the bottoming cycle. Work addition to produce high pressure steam is accomplished very efficiently by compressing water, after which waste heat from the MHD generator is used to turn the water into steam. In this particular application, the wave rotor acts as a form of heat pump, driven by a waste heat Rankine cycle. The peak outlet temperature of the wave rotor can be higher than that available from conventional compressors because of its inherent wall cooling characteristics. That is, the wave rotor walls are approximately at the mean of the steam and the air temperatures flowing through it.

Wave rotors have been developed for a similar application using high pressure helium to compress air to very high temperatures in the CAL Wave Superheater [8]. The wave superheater was a large wave rotor approximately 5 feet in diameter. It operated under extreme pressure and temperature conditions and could, therefore, only endure relatively short run times. The application envisaged for the MHD compressor could operate continuously at a somewhat lower temperature with essentially the same technology that was used on the CAL Superheater in the late 1960s. More recent design and materials improvement would provide an efficient wave rotor for this application.

The use of a wave rotor in a PFB power plant is to protect the turbine from the particulates in the hot combustion gases [9]. Usually several stages of hot gas cleanup are required before the gas can be expanded in the turbine to extract power. These cleanup stages are expensive, and they also reduce the work available in the combustion gas by dropping the pressure. Even after the gas cleanup, the particulate levels are still high enough to considerably shorten the blade life of the turbine. Part of the difficulty lies in the metal temperature of the turbines, which permits scouring by the particulates. Also, in order to extract work with a turbine, the gas velocity must change direction as it passes over the blades. The particulate velocity tends to lag these directional changes, causing particulates to strike the blade surface. In contrast, the wave rotor wall temperature will be lower than the turbine's under the same gas inlet conditions, implying a lower damage level. The wave rotor does not change the gas velocity direction, so that particulate also will not strike the walls as often. The net effect is a vastly reduced particulate erosion rate and an extended component lifetime. It also is possible to maintain or slightly increase the overall cycle efficiency by using the wave rotor, since some of the pressure drop associated with the cleanup loop can be eliminated.

In summary, the stationary power applications hold considerable promise, especially at higher cycle temperatures where the power plant efficiency will show the greatest gains. Both high efficiency wave rotors and high temperature, high pressure wave rotors have been demonstrated separately, but not yet in the same device. In the propulsion applications described below, the goals are different but the wave rotor requirements for combined high component efficiency and high temperature operation are much the same.

PROPULSION APPLICATIONS

There are two main reasons for using wave rotors for aircraft engines. The first reason - a relatively high component efficiency in a device that is automatically cooled at high temperatures - has been partly realized. The high temperature capability has been achieved with the high temperature materials available at each stage during the history of wave rotor development, and high component efficiency has been demonstrated (e.g., at MSNW) for low temperature operation. The absence of blade tip losses can maintain the wave rotor component efficiency, even for small engines, provided that leakage at the face of the rotor is minimized.

The second reason is the capability of very fast wave rotor response to external throttle conditions, allowing a wave rotor turbine engine to operate much closer to the compressor stall line. This benefit has been much harder to quantify because there still is no adequate model for the transient operation of wave rotors. However, its realization would mean that high performance engines can be upgraded for better cruise conditions, and engine reliability can be increased against inadvertant or sudden pressure fluctuations (e.g., at start-up or transition to dash).

Wave rotor turbine engine research was especially concentrated in the mid-1950s to late 1960s. During this period, aircraft turbine engines rose to prominence, achieving large advances in component efficiency and reliability. This feat was accomplished by metallurgical advances and by efficiency improvements. Those advances were made possible by designs which treated the turbines and compressors as steady flow systems, albeit with some important transient effects, such as compressor stall. The steady flow assumption vastly simplified the design problems and allowed much of the progress that occurred at that time.

The wave rotor, also in its infancy in the early 1950s, is patently an unsteady flow device. Consequently, it was much more difficult to understand and to design. Similar to other new technologies, there were a multitude of wave rotor configurations under consideration at that time, which compounded the problem of deciding which approach was the most appropriate for any individual application.

In retrospect, the historical choice to favor axial flow turbine and compressor research at the expense of wave rotor research seems obvious because the payoff was high and the perceived costs, in terms of relative design simplicity, were lower.

Recent advances make it worthwhile to reconsider the use of wave rotors for aircraft turbine engines. These advances include improvements in computational fluid dynamics, the acquisition of critical new experimental data on wave rotor performance, and the fact that advanced turbine engine performance is constrained more than ever by high temperature materials development.

One of these advances involves the development of computer flow simulation codes, which allow very accurate prediction of wave rotor performance. One of these has been verified by detailed data taken at the MSNW wave rotor test facility. This code has been used to speed up modifications of the original wave rotor designs to make them more efficient. That efficiency increase has been measured in subsequent wave rotor tests. The capabilities of this vastly improved computational technique overcome one of the primary causes of failure (i.e., slow, burdensome design optimization) in earlier efforts.

The Comprex development by Brown-Boveri also has established the fabrication techniques for wave rotors in commercial quantities. With the fast computational design codes and the Brown-Boveri technological advances for wave rotors, it is time to look again at applying these devices to those aircraft applications where they may have their greatest advantage; namely, to small, low cross section, long-range engines and to high maneuverability engines.

THE KLAPPROTH ROTOR

Over a period of 8 to 10 years, General Electric (GE) considered several wave rotor configurations in the context of advanced turbine engines. The engines considered were primarily turbo-props. They carried out an extensive experimental and analytical program. Both pure pressure exchange and shaft work output wave rotors were investigated. The former took the form of a device very similar to the modern Comprex device; it had straight tubes parallel to the rotor axle. The second device, which we have dubbed the Klapproth rotor since it survives to this day and was associated with Klapproth's efforts in the GE research program, has helical tubes which are capable of receiving impulsive thrust from the inlet gas flows and which can deliver reactive thrust from exit gas azimuthal velocities different from the rotor tip speed.

It appears that each of these rotors succeeded to a certain extent; namely, to validate the compression and expansion processes occurring with each cycle on the rotor. But the Klapproth rotor in particular apparently never succeeded in producing significant shaft work output. In this mode it would have operated as a compressor-turbine combination with a combustor to add energy to the compressed air flow.

At that time it took a gas dynamicist many days to generate a wave diagram describing the internal gas flows for a wave rotor. This was accomplished via painstaking hand calculations using the method of characteristics. Even the smallest design change would require a recalculation, with a delay while the new flows were calculated. These calculations were supplemented by some excellent and complex water table experiments which could simulate some of the time-dependent and design-dependent aspects of wave rotors. However, there was a basic lack of empirical data and a lack of understanding of how these devices actually behaved because of inadequate modeling.

For example, throughout this period (and even to this day) there has been a temptation to assume that the waves could be drawn with zero width, as if they were generated at a discrete point in time and space and did not spread or deform later in time. Little recognition was given to the need to account for several internal reflections of these waves in order to properly calculate the flow magnitudes. One of the most important inconsistencies which appeared in early studies was that steady flows in the manifolds external to the wave rotors could be achieved without also requiring that the wave patterns used in the design be periodic with each revolution. As a result of this last assumption, the design calculations generally mis-estimated the actual performance by a substantial amount.

When each of these design requirements is taken into account, there is good reason why some of the initial rotors did not always perform satisfactorily. With the aid of modern computational techniques, the design, for example, of the Klapproth rotor, can be improved.

THE ROLLS-ROYCE ROTOR

Rolls-Royce (RR) has carried on an experimental wave rotor test program and a parallel analytical effort. Consultants to Brown-Boveri assisted RR in this latter effort, carrying out some of the numerical calculations required to generate the wave diagrams and rotor designs. As in the GE case, these calculations were performed by hand and each design took several weeks to complete. The RR program began in the 1960s and perhaps lasted as late as 1972, when the company went into receivership and was taken over by the government. Clearly, the loss of revenue at that point stopped various research efforts, some of them successful and some of them relatively high risk. The wave rotor program was no exception; however, they were successful in closely approximating their design goals before the program was cancelled.

While available data is sparse, the Rolls-Royce rotor(s) were evidently pressure exchange wave rotors; in fact, at least one of them was related to a Brown-Boveri Comprex with similar manifolds (though different placement) to those used with the Diesel supercharger. The application appears to have been to a gas turbine cycle with a wave rotor high pressure stage as a topping cycle, similar to that described by Berchtold and Lutz [10] in their papers. Test data was taken to verify the performance of this device. This data included mass and energy flows for each of the manifolds and some measure of the gas leakage, which was difficult to control and, from their viewpoint, precluded the achievement of the pressure ratio they were looking for. In a later development, they were able to reconcile the predicted performance with measured data.

To conclude, the RR device operated nearly according to their predictions. However, they also suffered problems regarding the difficulty in understanding the gas flows and in developing optimized designs for their wave rotors. They identified leakage as a significant mechanical design problem in their experiments.

It would be very valuable to be able to scrutinize more of the RR wave rotor experience. Details on the actual performance measured and their assessment of technology problems would add to our understanding of what needs to be done to make a successful wave rotor. In balance, they achieved definite, positive performance results.

THE PEARSON ROTOR

In the mid-1950s, the Ruston-Hornsby Turbine Company supported the construction and testing of a different kind of wave rotor designed by Ronald Pearson. The blades of this device had long helical sections and cambered ends. That is, the blades or tube walls were bent somewhat at each end, which changed the direction of the gas flows more like a conventional turbine blade. However, the Pearson rotor was very much a wave rotor since part of the cycle was devoted to compressing the inlet air while the rest of the rotor cycle utilized the expansion of the exhausting gas (heated by an external combustor) to extract shaft work.

Based on experience with a smaller prototype rotor, the Pearson wave engine was designed and built in less than a year, and it produced shaft power from the first moment it was fired up. In this regard, the Pearson rotor was singularly successful. It operated over a relatively wide range of off-design conditions, producing net shaft work in the range of 5 to 35 horsepower, which was very close to its projected operation.

While Pearson also had to perform many tedious wave diagram calculations by hand, he had adopted a more modern design philosophy from the start. First, he recognized that the wave system must be periodic and enforced that condition in all of his calculations. Second, he accounted for all of the internal wave reflections and, in many instances, designed extra ports whose purpose it was to control

and/or cancel these reflected waves. He also took a fundamental approach to guarantee that his wave diagrams would produce net work output; that is, the pressure, mass flow, and temperature conditions imposed as boundary conditions on the wave diagram calculations were prescribed in such a way as to produce net work.

The Pearson rotor was advanced for its time in terms of using abradable seals which could be held to close clearances and in the brazed construction of the rather convoluted rotor tubes. The bearings required special attention, and the thermal expansion of the rotor components was very carefully considered.

The wave rotor experiments were clearly outside the norm for projects at Ruston-Hornsby and were promoted by one of the directors of the company who took a special interest in Pearson's device. As a result, the project was considered expendable and, when it suffered a setback due to overspeeding from an improperly connected fuel line, the project was cancelled and Ruston returned to its customary product development.

The Pearson rotor is a very important piece of evidence supporting the idea that wave rotors can be made to work effectively. Its success refutes the claim that such devices only work over a very narrow operating range. The Pearson design contains specific ports and nozzle vanes whose sole purpose was to maintain high component efficiency over a very wide range of operating conditions. These design details were tested and improved upon during the project at Ruston-Hornsby; the experimental data supported the worth of each of these remedies. The only other rotor including the capability for good off-design performance is the Brown-Boveri Comprex, with closed cavities or "pockets" at points around the periphery of the endwalls which control wave reflections at those walls. The Comprex experience and data also support the capability of wave rotors acting as pure pressure exchangers to operate over a wide range of conditions, especially at part load.

THE GPC ROTOR

The General Power Corporation (GPC) is one of the few firms to have maintained some degree of continuity in a wave rotor research program, having begun in the mid-1960s and still being active now. The GPC rotor shares some of the features of the Klapproth rotor and of the Pearson rotor, but there the resemblance ceases once one considers the details of its design and operation. Its purpose is to generate shaft horsepower, and it utilizes helical tubes with a bend (camber) near the outlet to change the direction of the gas flow on the rotor. The main part of each tube has a stagger angle much like the Klapproth rotor (i.e., helical blades). In the early GPC patent disclosures, the wave diagrams were described with infinitesimally thin waves, with no spreading in time.

The GPC rotor development was originally intended for a road vehicle engine, which would require a relatively high pressure ratio Brayton cycle with several expansion stages on the wave rotor. The curved blades were intended to act as a reaction turbine for this portion of the flow in order to produce net shaft output power. Each stage was fed by the preceding exhaust stage gas with a re-entrant duct which curled around from one side of the rotor to the other.~ Work on this device has been supported first by the Ford Motor Company and later by the Department of Energy and DARPA.

A simplified version of the GPC rotor, which includes just one re-entrant duct for a lower overall pressure ratio, has been tested recently. However, the basic rotor design has not changed significantly. To date, this rotor has not produced any significant net output power. An initial inspection of its design suggests that the tube curvature may be too large; that is, too much reaction may be encountered during the expansion stages, robbing the exhaust flows of the pressure required to re-inject the flow into the next expansion stage. Hence, one would suspect poor scavenging under these conditions would result in a loss of available output power.

Unfortunately, the GPC work is poorly documented at present, making it difficult to draw any certain conclusions about its performance. When contrasted to the Pearson device, there are several areas of the GPC design which may cause problems. The first, discussed above, concerns the degree of bending in the tubes. The second concerns the lack of control over reflected waves within the device required to make the wave system periodic within one revolution. The third area is the absence of any strong impulsive loading of the rotor from inlet manifolds to produce shaft work; the Pearson device relies heavily on impulsive loading to achieve power output and has only a very mild change in the blade angle to produce a small amount of reactive power.

Any one of these design problems would be sufficient to cause severe problems in power output from a wave rotor/turbine. The GPC difficulties were compounded in the earlier years by the same difficulties in computing the wave patterns which other researchers had. They have recently developed a characteristics computer code which is potentially accurate and fast but involves a very complex accounting system of characteristics nodes in order to achieve high accuracy; each time a wave reflects within the system, it adds to the number of such nodes which must be accounted for and multiplies the length of time that such a computation must run to get good results. Even so, such a technique will cut many hours from the normal design calculation done by hand.

The GPC approach began with a rotor design that was destined to be a complete vehicle engine, so they were initially encumbered with a multitude of design problems which were somewhat extraneous to the task of learning how the wave processes would evolve on their device. Some of those design problems have been solved, but the GPC rotor is still a difficult system for learning about or verifying wave rotor performance.

THE COMPREX

The Brown-Boveri Company has been involved with wave rotors since the beginning of this technology. Their staff and consultants have investigated a number of different applications including the use of wave rotors as topping stages in a turbine engine. Ultimately, they settled on the development of the Comprex as offering the maximum payoff consistent with their corporate objectives. The Comprex is a supercharger for Diesel engines. This device has tubes parallel to the rotor axle and is designed to have a uniform performance over a wide range of road conditions. The Comprex is a commercial device and has been successfully tested on a wide variety of vehicles such as the Mercedes-Benz Diesel car, the Peugeot, and Ferrari; on Diesel trucks manufactured by M.A.N.; and on bulldozers. A great deal of development has been applied to making the Comprex a durable device which meets the stringent conditions of modern automobile engines; currently it is sold as the factory-installed supercharger on a class of European-produced tractors.

The Comprex does not have to be very efficient in the supercharger application because there is generally surplus available work in the engine exhaust stream over and above that needed to compress the incoming air (i.e., to supercharge the engine). At

good part-load performance. The Comprex does not attempt all of the refinements also needed to achieve high efficiency; for example, those found in the Pearson device.

The Comprex is one of the most important examples of successful wave rotor technology. It is a commercial device. Furthermore, the Brown-Boveri Company has solved the most difficult development problems for their application – the seals problem and the thermal stress problem. Leakage is kept to an acceptable level in the Comprex by enclosing the rotor in a pressurized shroud and by using a rotor material with a low thermal expansion coefficient over the range of temperatures suitable to the Comprex. The thermal stresses are managed by routing the hot gases in and out the same end of the rotor and the cold gases in and out of the other end. Also, a "cathedral" design using an alternating arch cross-sectional shape for the tubes helps to allow thermal growth in the tube walls without placing large stresses on the outer rim of the rotor.

WAVE ROTOR TURBOMACHINES FOR THE 1980s

As we have seen, the past history of wave rotor development had mixed levels of achievement. The reasons for this are appreciated sufficiently so that a realistic appraisal of wave rotor technology as a way to improve the performance of present-day turbine engines is now possible.

During the last ten years, turbine engine performance has been limited in part by the peak turbine inlet temperatures allowed by turbine blade materials and by blade cooling. Blade cooling puts an extra drain on the compressor and begins to reach a point of diminishing returns when the temperature difference between the inlet and the blades increases to a few hundred degrees. Thus, the burden for further improvement, at least in the core engine, falls back on high temperature materials development. Significant improvements have been made over the past decade on hot stage turbine materials; reliable performance now can be achieved with inlet temperatures of 1900 to 2000°F, and for limited durations up to as high as 2500°F without blade cooling. With blade cooling, these limits can be extended upwards by a few hundred degrees.

The wave rotor automatically cools itself without diverting any gas streams from the compression part of the cycle. Because the cooler input air is periodically cycled on and off of the rotor, the rotor wall sees both the cold and the hot gas streams and assumes an intermediate temperature between these two. The actual rotor wall temperature depends on the details of the heat transfer with these two gas streams but is generally much lower than the inlet combustion gas temperature. Further, no special passages have to be drilled into the walls of the wave rotor in order for it to be cooled. The cooling is accomplished within the main gas passages consisting of the compression and expansion tubes surrounding the rotor. As a consequence, cooling in small wave rotors is achieved with ease compared to the problems of blade cooling in small turbines. Since the wall temperature is somewhere near the mean of all of the inlet and outlet gas temperatures, the rotor temperature may differ by considerably more than a few hundred degrees from the peak gas temperature, implying that higher peak gas temperatures may be used

low part load, any deficiency in available work is rapidly compensated for by the ability of the Comprex to rise quickly to full-load performance. Ordinary turbochargers suffer some lag in this respect because of their rotational inertia. As a result, the current Comprex designs emphasize a broad performance range. As mentioned earlier, pockets are cast into the rotor endwalls to control wave reflections and to achieve

with this device than with a cooled gas turbine. Of course, any material improvements available for advanced turbines also may be used to boost the temperature capabilities of the wave rotor.

In essence, the wave rotor should be considered as a high temperature, high pressure stage in a turbine engine because it will protect the power extraction turbine from the peak gas temperatures of the combustor. Its high temperature capabilities should be used to boost the cycle efficiency (and lower the thrust-specific fuel consumption) of present-day gas turbine engines and to eliminate the need of entering a difficult materials development program for the more advanced turbine engines. By keeping the high tip speed turbine components at a lower temperature, the wave rotor also will improve the overall reliability of these engines.

The chief question underlying these suggestions to reconsider the wave rotor is no longer whether or not it will work. Instead, one needs to demonstrate how well these devices will work, since their component efficiencies at higher temperatures will be compromised by the amount of heat transfer and gas leakage which accompanies high temperature applications. The initial estimates of these effects indicated that the component efficiency can be maintained at a high value. Also, since the wave rotor is being considered primarily as the high pressure stage, the overall engine performance is not particularly sensitive to the efficiency of that stage; that is, any losses from work transfer or shaft work production in the top stage are available to do work in subsequent stages, although not with the same effectiveness as if that work were accomplished in the top stage. This cascade effect is well documented in combined cycle operation, where the topping cycle is often rather inefficient but chosen simply for its ability to operate at higher temperatures (for example, the MHD-steam turbine combined cycle power system) and to thereby boost the overall thermodynamic cycle efficiency.

Besides its advantages of cool walls and good efficiency in small sizes, wave rotors appear to behave in a substantially different way to pressure transients in an engine configuration, compared to a conventional turbine engine. One of the principal limits to high performance aircraft engines is the surge line marking the beginning of compressor stall. If the throttle is opened too quickly by the engine operator, the pressure rises in the combustor and creates a temporary decrease in the equivalent mass flow exiting from the compressor. Because of the inertial masses of the compressor-turbine pair, the compressor maintains roughly constant rotational speed during this transient, moving the compressor performance across the surge line where it will stall, with an immediate loss in power. Thus, the operator must not accelerate the engine too rapidly. Military engines are derated below their performance potential so that an adequate safety margin is provided for acceleration without compressor stall.

The transient response of the wave rotor is extremely fast because the wave patterns internal to the device can readjust on timescales like the acoustic transit time along the length of the rotor, e.g., on the order of a millisecond. The rotational speed of the rotor is not the primary variable in this instance since it controls the valving of the gas on and off of the rotor but not the mass flow or pressure constraints of the system. As a result of its fast response time and because it is placed between the combustor and the compressor, a wave rotor will allow an aircraft turbine engine to operate much closer to the compressor surge line without any danger of compressor stall. The implications of this change are

that the overall performance of the turbine engine can be increased under ordinary flight conditions and it will be effectively stable to either accidental or intentional changes in the combustor (i.e., throttle) conditions on very short timescales. Thus, a wave rotor turbine engine can be very reliable since it will not accidentally stall and it can be programmed or driven to very rapid maneuvers. Such an engine can be operated under cruise conditions closer to the stall line and, hence, at higher efficiency and lower thrust-specific fuel consumption to obtain increases in range; for unanticipated situations (an aborted landing, for example), the pilot would have the capability of pulling out (i.e., accelerating) with a much smaller margin than with a conventional engine.

The transient response of the wave rotor has been demonstrated during the test series performed by Ruston-Hornsby on the Pearson rotor. There is every reason to believe that this is a general property of such devices but, of course, it should and can be confirmed by analysis and experiments for any given wave rotor design.

SUMMARY

The history of wave rotor turbine engines shows that two such engines have been built and demonstrated in the laboratory close to their projected performance over a wide range of operating conditions. These are the Pearson rotor and the Rolls-Royce rotor (alias the Brown-Boveri Comprex). Their success appears to be due to a concerted effort to understand the internal wave processes and to incorporate wave control remedies in the rotor design. These remedies also were aimed at providing a very wide range of operating conditions for these rotors without undue fall-off in performance. The Pearson device, in particular, achieved net work output by also understanding the fundamental trade-off between reactive forces and impulsive forces and the need for closed loop gas scavenging on the rotor. New computer design techniques verified by recent experimental data now allow rapid design optimization of wave rotors so that a successful development program for these devices could be completed in a relatively short period of time.

The development of advanced turbine engines is pushing the limits of high temperature materials. Wave rotors would boost the high temperature limits to turbines, allowing better and more reliable performance, especially in small-size engines where cooling and component efficiency are difficult to achieve. No new materials development would be required for the wave rotor turbine engine to yield significant improvement over existing turbines. If higher temperature materials become available in the future, they would simply increase these improvements.

Both larger and smaller wave rotor turbine engines would benefit from the wave rotor's ability to stabilize engine transients during sudden changes in the throttle conditions, allowing better cruise performance, maneuverability, and reliability.

A critical need exists at this point which is not being addressed by any R & D program; namely, a detailed experimental verification of the performance projected for hot stage versions of either the wave rotor/turbine or the pressure exchange wave rotor or, preferably, both. To advance beyond what Pearson and Ruston-Hornsby learned 25 years ago, it is necessary to incorporate improvements in the wave rotor design and to run well-instrumented tests on its performance in the design range of operating conditions required for a particular application. The chief requirements in this regard are to measure and control the leakage and to design for a higher peak temperature than was considered at that time. The knowledge gained from the DOE and DARPA wave rotor programs has provided the basis for the design of an efficient wave rotor.

The pressure exchange wave rotor may offer a lower risk, faster development alternative. The pressure exchanger version of the wave rotor typically does not have to operate at as high a tip speed as the wave rotor/turbine because it does not have to meet propulsive efficiency requirements associated with the high speed of sound of the combustion gases. It also may be more efficient than the wave rotor/turbine. The combination of lower tip speed and higher efficiency means it would be easier to develop a highly reliable wave rotor for a given peak cycle temperature. There may be some penalties to this approach in the complexity of the resulting engine since an extra, rotating component beyond the turbine and compressor is required. A careful design of the turbine engine configurations resulting from these two wave rotor approaches needs to be carried out to answer this question.

In short, wave rotors have been operated successfully in the laboratory in the past. Those designs can be improved upon quickly to upgrade their efficiency and performance. Well-instrumented tests of a wave rotor at higher temperatures and pressures would establish the feasibility of this technology and confirm the validity of the existing design tools which would be needed for an engine development program.

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LECTURE SERIES 1982-01

VON KARMAN INSTITUTE FOR FLUID DYNAMICS
CHAUSSEE DE WATERLOO, 72
B - 1640 RHODE SAINT GENESE, BELGIUM

TURBOCHARGERS AND RELATED PROBLEMS

LECTURE SERIES DIRECTORS : F.S. BHINDER
R. VAN DEN BRAEMBUSSCHE

22 pages

JANUARY 11 - 15, 1982

SUPERCHARGING WITH COMPREX

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Zürich / Switzerland

History

The idea of interchange of energy by direct impingement of gases dates back to the time at the beginning of this century when Büchi proposed exhaust gas driven turbosupercharging of Diesel engines. A patent of Burghard describes the cell rotor admitting gases through segmented stator ports. The idea forgotten came up again in the late thirties when the principle was used as a heat pump with air as the working fluid. Brown Boveri who was building the device for a customer realized that design modifications were needed to account for nonsteady flow phenomena. The patent by C. Seippel describes the operation and the timing requirements. Later, the pressure wave energy exchanger named Comprex to indicate its capability of compressor-expander was used as the upper stage of a gasturbine engine. Whereas good efficiencies were obtained, rotors could not be built to stay together. In the early fifties, the potential of the Comprex as a Diesel engine supercharger was recognized. After the first units were constructed and tested on vehicular Diesel engines in the USA, Brown Boveri in Switzerland initiated a concentrated effort to develop the Comprex to meet the requirements of its practical realization.

The Principle of Operation

The schematic arrangement of the Comprex as exhaust gas driven Diesel engine supercharger is shown in Fig. 1. The rotor B has straight axial channels of constant cross section, also called cells. The stators on both sides of the rotor have segmented openings registering with the rotor passages. The ducting F for air entering the rotor and for the discharge of compressed air E are connected to the cold stator. The duct for compressed hot gas D and the duct for hot gas discharge G are connected to the hot stator. The rotor assumes essentially a valving function. The belt drive C is needed to overcome bearing-, windage- and friction-losses. The energy exchange comes from direct interaction of the gas to be expanded, passed on to the air to be compressed.

The mode of operation can best be explained considering the axial rotor channels to be unwrapped into the projection plane Fig. 2. The ports in the cold stator on the left side and the ports in the hot stator on the right side are stationary. The duct E for the compressed air connects to the en-

gine intake manifold E. The duct D for the exhaust gas collected in the exhaust manifold A leads into the rotor. The turning of the rotor is represented by the rotor cells moving from the top on down. The ports thus open and close according to the schedule determined by the geometrical location of the ports and the downward speed of the rotor. The requirements for the timing of the compression- and expansion-waves are best understood by the description of the cycle.

The explanation begins on top of Fig. 2 where the rotor is filled with air at rest below ambient pressure. The cell opening to the exhaust gas port allows the gas to enter the rotor due to the large pressure difference. The air at rest has to be pushed forward producing a compression wave. The wave front progressing faster than the speed of sound simultaneously compresses and accelerates the air. As the wave reaches the cold stator, the discharge port is opened and allows the compressed air to enter the stationary duct E leading to the engine intake manifold E. The kinetic energy is partly converted int pressure. The speed of the gas and the speed of the compressed air are at it equal. This also is the speed of the interface. As Fig. 2 indicates, this speed is far below the speed of the compression wave. It is this physical phenomenon which makes the efficient transfer of energy by direct interaction possible.

The compressed air discharge port closes prior to the arrival of the interface preventing exhaust gas to be recirculated into the engine. Some of the air which has been contaminated due to mixing at the interface remains in the rotor cell. As the exhaust gas intake port is closed a rarefaction or expansion wave is generated. The deceleration of the gas to standstill at the wave front occurs with the simultaneous expansion. Now the rotor is filled with exhaust gas and some air, both at rest. The pressure is still considerably above ambient. This remaining pressure furnishes the energy to scavenge the cell, namely to replace the exhaust gas by air. The opening of the exhaust gas discharge port induces the second rarefaction or expansion wave, thus generating the outflow into the exhaust duct G. First the gas and then the air in contact is being traversed by the expansion wave. The full content of the rotor cell is set in motion. At this time the air intake opens and the inertia of the gas column draws in the air. As the cell has been scavenged the exhaust discharge port is being closed. Prior to this time the air intake has also been closed, thus producing another, however rather weak expansion wave decelerating the air to standstill. The pressure of the air at rest is subambient. The cell is now ready for the next cycle.

The pressure of the air obtained in the intake manifold E is determined by the strength of the compression wave which again is a function of the engine exhaust discharge pressure. This again depends on the exhaust gas temperature leaving the engine. As the mass-flows to and from the engine are essentially equal, the power to be gained by the expansion of the hot exhaust gas is sufficient to obtain a somewhat higher pressure of the cold air to be compressed. Part of the surplus power lowers the inefficiencies. There are losses due to the finite clearance between rotor and stators. Furthermore, there are losses due to incomplete recovery of kinetic energy of the compressed air in the duct E and of the exhaust gases leaving the rotor through duct G. Losses are caused also during opening and closing of the cells passing the radial edges of the stator ports. The effect of heat exchange superimposed to the compression and expansion energy exchange makes the processes non-adiabatic. This is a distinct difference from the adiabatic compression and expansion in turbomachines. Work dissipation due to flow friction on the cell walls brings a further deviation from the isentropic changes of state. The losses due to irreversibilities in the compression wave are insignificant. The design point including

the total of all losses reveals overall combined energy efficiencies of 74 %. This makes the Comprex competitive with turbosuperchargers.

Matching the Comprex to Engine Demands

A practical supercharger has to function over a wide operating range. Vehicular engines in particular are rather challenging in this respect.

The rotor as shown above has to be driven by the engine. V belts are well accepted on vehicular engines for generator-air-fan drives. This, however, makes the Comprex speed range as wide as the engine speed range, namely up to 1:4. At low Comprex speed the main compression wave arrives far too early at the cold stator with the result that the compression wave is being reflected at the closed end of the rotor cell creating a high pressure of no use. This wave returning to the exhaust gas intake disrupts the timing with the effect of flow reversal and with a complete breakdown of performance. A number of simple means have been found which help to overcome the disadvantage of wave mistiming. The wide speed operation has been realized by the use of additional stator ports which, however, are not connected to any duct. The ports allow the air in the rotor to flow out and to enter the rotor into the adjacent cell. In the case of the early arrival of the compression wave, only partial reflexion occurs. The reflected wave arrives at the hot stator at the time the inflow of exhaust gas into the rotor has started. The wave prevents the inflow over a segment of the port. The flow of exhaust gas is reduced which matches the lower flow at reduced engine speed. This appears to the engine to have the same effect as variable geometry of a turbocharger. A similar pocket is supplied with exhaust gas. The secondary wave generated by this pocket enters into a corresponding pocket in the cold stator. The effect of these pockets sustains scavenging of the rotor cells at low rotor speeds. In Fig. 3 the essential components of a Comprex unit are shown. The main ports and the axiliary ports (as pockets) are visible. The letters indicating the ports agree with the duct designation of Fig. 1. The letters HIK refer to the pockets. The stators, contrary to Fig. 1 and 2, are built to produce two complete cycles per revolution of the rotor. This has the advantage of symmetry in the stator design. At equal rotor speed the rotor length is half which makes the unit lighter and more compact.

The Comprex having a porting configuration to give a wide speed range fortunately meets the massflow demand of the engine. This results in a nearly constant engine intake manifold pressure for constant exhaust gas temperature in a reasonably wide speed range of the engine.

The requirements for passenger car- and truck-engines as well as the capability of rapid load changes are being discussed in the second part of the paper.

In the supercharger version the peak efficiency has to be sacrificed in favour of the wide speed range capacity. Yet the efficiencies are competitive with turbosuperchargers which need bypass blowoff valves causing a considerable loss in efficiency at high engine speed.
The Comprex can be analysed theoretically using the method of characteristics. The assumption of onedimensional flow can be refined to take care of partial opening losses, leakage losses, as well as heattransfer- and friction-losses. Integration of the theoretical intake- and exhaust velocity profiles gives remarkably precise duplication of measured performance. The cycle calculation using the graphical method of characteristics is rather a demanding and time consuming process. Whereas it is possible to computerize the calculations, it is still difficult to introduce complicated boundary condi-

tions such as pockets since massbalances will have to be fulfilled for the main ports as well as for the pockets. The graphical method of analysis can be made with grossly simplified assumptions. It is most instructive since it allows to understand the phenomena which determine basically the functioning of the Comprex.

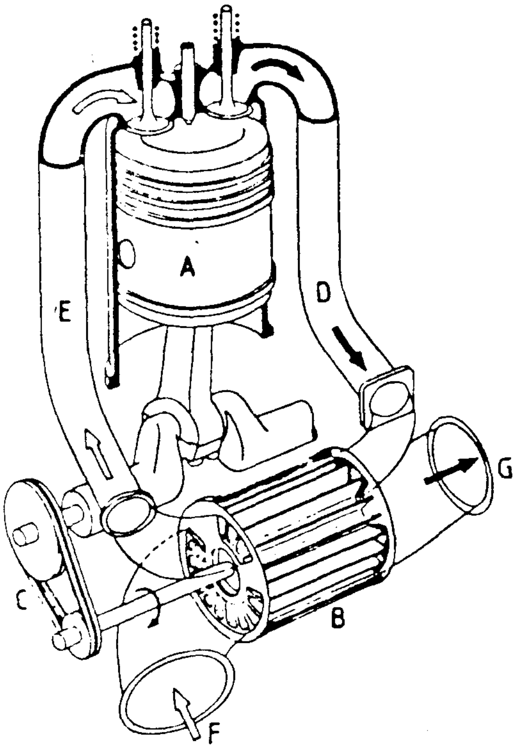

Fig.1. COMPREX Pressure-wave Machine
as a supercharger
A Engine
B Cell-wheel
C Belt drive
D High-pressure gas
E High-pressure air
F Low-pressure air
G Low-pressure gas

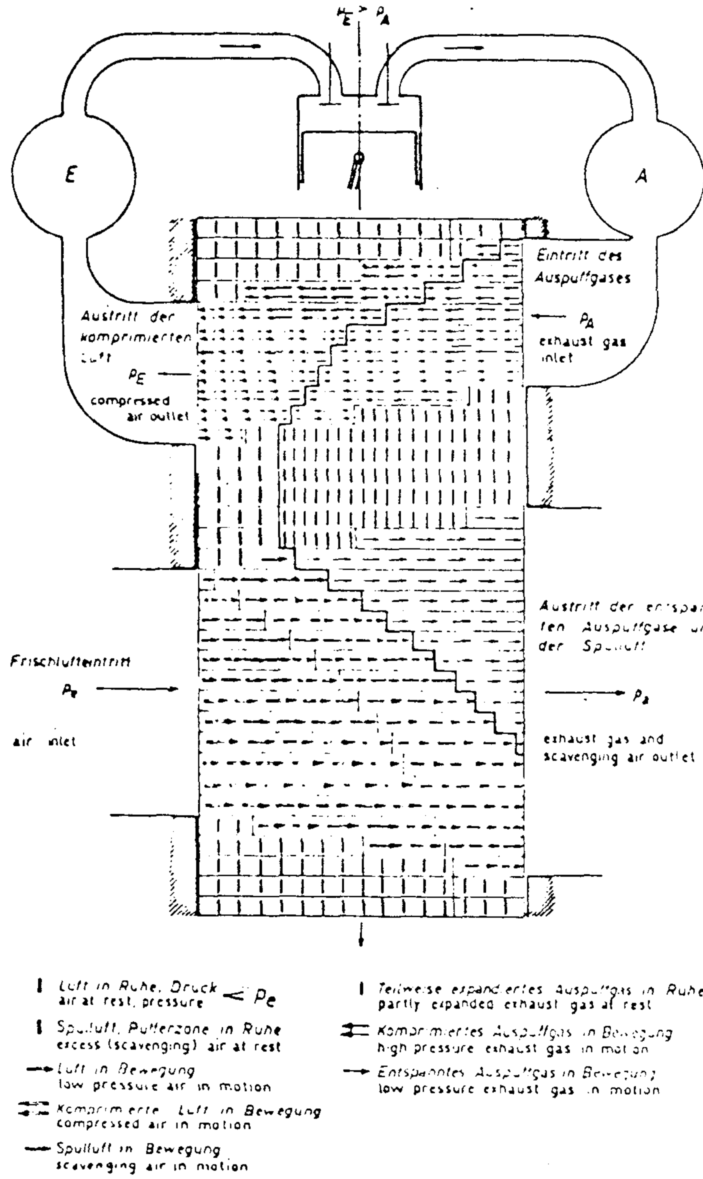

Fig.2. Unwrapped Rotor Cells
Wave Propagation and
Air-resp. Gas-flow schedule.

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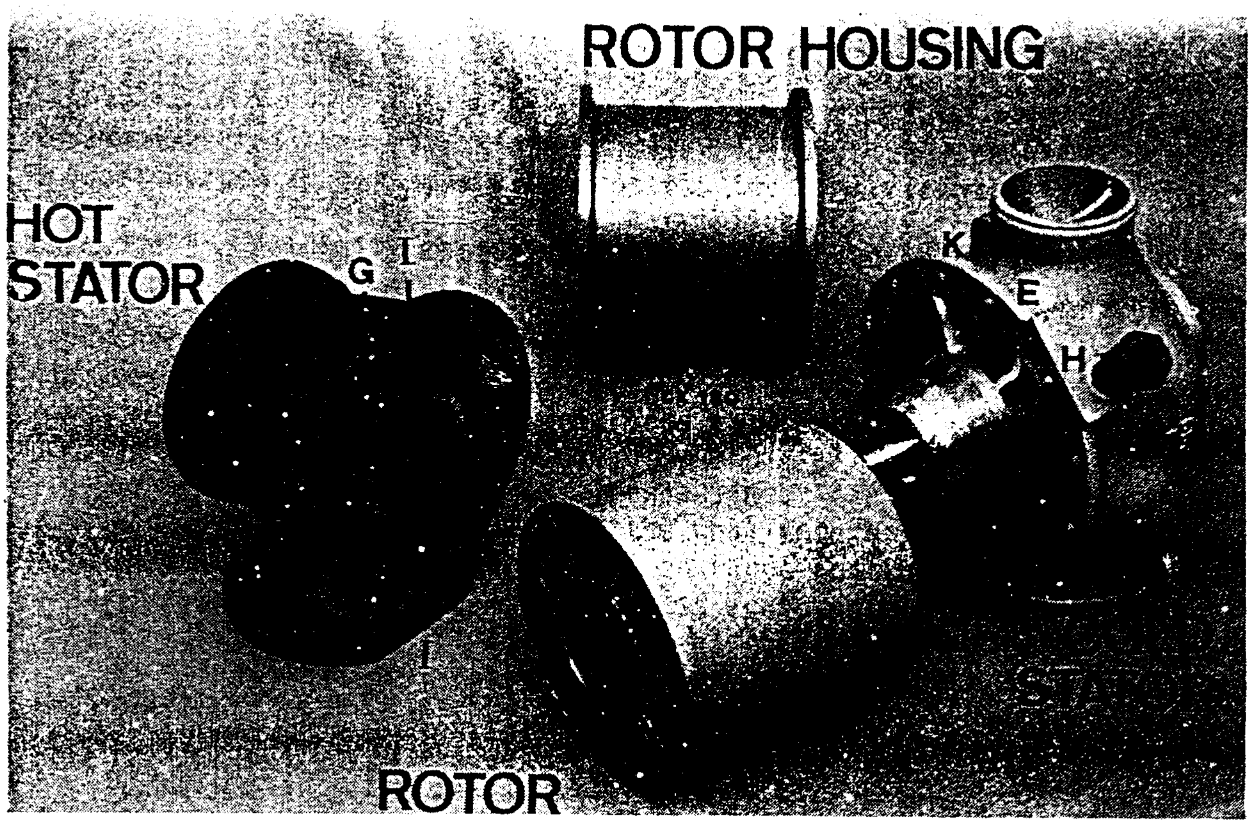

Fig.3. Components of the Comprex.
Ports D,G,E,F are connected to Ducts D,G,E,F Fig.1.
D Exhaust gas from engine
G Exhaust gas discharge
F Air inflow from ambient
E Compressed air flow to engine
Pockets H,I,K are energized at low rotor speed and provide energy for scavenging the cells.

SUPERCHARGING WITH COMPREX. APPLICATION AND EXPERIENCE

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Introduction

Today almost all Diesel engines of power output higher than 400 kW are turbosupercharged. Brown Boveri has contributed significantly in raising the level of supercharging technology. A triplication of the power output compared to the naturally aspirated engine of equal displacement has become an accepted standard.

Supercharging is broadly applied to Diesel engines for trucks, buses, farm machinery and earth moving equipment. Here the operational requirements are more demanding due to the frequent load changes and the wide engine speed range. The reason for the wide use of the Diesel engine is the high fuel economy and the long durability. The passenger car Diesel which has become more widely considered has to meet the challenge of high fuel economy. The requirements for the wider engine speed range and the instantaneous load respone are well recognized. Noise levels, acceptable for trucks, have to be reduced substantially. The emission levels reached in spark ignited engines only with sophisticated exhaust gas after treatment, are met by the Diesel

engine with respect to carbon dioxyd and hydrocarbons. Nitric oxyd and particulates, however, as well as the unpleasant odour are of primary concern.

The low Diesel engine power output compared to gasoline engines of equal weight makes supercharging mandatory. The Comprex pressure-wave supercharger has proven its capability to meet these requirements of the passenger car Diesel engine.

Tests of supercharging Diesel engines with the Comprex go back over 20 years. The first prototype units have been tested and evaluated in a vehicular application in 1969 on a heavy duty truck, built by the Saurer Company in Switzerland. The performance expectations have been met. A number of problems, however, had to be solved to make this new device suitable for practical use and to become competitive in price with the conventional exhaust gas driven turbocompressor.

Manufacturing Technology

In order to meet durability at low cost, a rotor casting process had to be developed. A tremendous effort led to a sophisticated technique, capable of producing rotors which have numerous advantages over the older rotor construction, using sheet metal blades brazed into slots in the hub and the shroud. The casting technique allows to form the blade joints at the hub and the tip without local stress concentration. The intermediate shroud permits the stresses due to different temperatures between the hub and the shroud to be relieved. Previously this has been obtained by curved blades in radial direction. The alloy for the rotors is of a low coefficient of thermal expansion. Furthermore, it has a high oxydation resistance. Operating gas temperatures up to 850° C can readily be handled. The casting facility is shown in Fig. 1.

Precision casting techniques also are being applied to the stator castings. The porting geometry meets the tolerance requirements without expensive machining operations.

The Comprex rpm being between 8000 and 25000 depending on the airflow capacity, presents no problem to bearing design. The bearings are supplied by the engine oil pump. Rubbing seals are used to prevent oil from entering the air passages or dirt to get into the oil return line. The partially segmented Comprex, Fig. 2, shows the shaft and bearing arrangement.

Refinements in the stator design have resulted in a substantial weight reduction. The unit is supported by the exhaust gas collector, an arrangement well proven by the mounting of turbo superchargers.

Noise emission

The previous sheet metal rotor produced a narrow frequencyband noise of a rather penetrating character. The effective dampening to a permissible level turned out to be difficult. Cells of uneven width in random pattern gave a significant reduction in noise. The cast rotor with the intermediate shroud in variable cell width and shifted cell partitions also for reasons of stress relief gave an even greater noise reduction of 10 db. The additional means for treatment of the remaining noise can thus be less effective. The noise emission spectra for the two different rotors is shown in Fig. 3.

Exhaust Gas Recirculation

Diesel engines operating at low bmep produce NO_x even though the mean tempe-

rature is far below the stoechiometric combustion end temperature. The burning, however, always takes place at the zone of stoechiometric mixture. The local burning temperature therefore is as high as at full load. A reduction of the reaction heat thus calls for a reduction of oxygen concentration. This is obtained by exhaust gas recirculation, reducing the stoechiometric peak cycle temperature determining the NO_x equilibrium. A simple valving- and control system is used to have some exhaust gas bypassed into the engine at part load. At full load, the recirculation which causes a reduction of power output is undesired. Fortunately, the Comprex by its principle of operation has a sufficient amount of internal exhaust gas recirculation in the particular desired engine operating regime. Fig. 4 indicates the amounts of recirculation for different operating conditions. The NO_x reduction comes to 20 % at low load.

Potential Power Output Gain by Supercharging

The requirements for the supercharging of Diesel engines depend on the application. Passenger car engines are the most demanding. The high torque at low engine speed, combined with the request of rapid load response is typical. Whereas the turbosupercharger, combined with a turbine bypass valve, is capable of building up a sufficient manifold pressure at low speed, it is not giving the response. At high engine speed, the limited turbine flow capacity calls for opening the bypass in order to prevent the manifold pressure from exeeding permissible limits. The compressor stability, however, remains to be a problem.

In the case of a more narrow speed range, such as in traction engines with multiple speed gearboxes, the waste gate hookup can be used to a pressure ratio level of 3. About the same limitations exist for the Comprex as far as speed range and pressure ratio. Fig. 5 shows the ranges for passenger car and truck engines.

As special engine applications call for higher levels of supercharging, two stage supercharging appears to be the best solution to meet the vehicular requirements. It has been found that the combination of the Comprex as the low pressure stage and a turbocharger as the high stage offers significant advantages both with respect to high speed range and high response. A peak bmep level of 27 bar ata at a manifold pressure ratio up to 6 has been demonstrated. The tests have confirmed the performance expectations. A six cylinder Caterpillar engine has been used which has been equipped with a fuel pump of higher capacity and with shorter pistons reducing the compression ratio to 12. Furthermore, it became necessary to install a device for adjustment of the injection timing.

Cooling of the charge air lowers the peak cycle pressure in all cases and thus the thermal loading of the critical engine components.

Comprex Superchargers for Wide Range of Engine Capacity

At the first stage of development, a series of Comprex units has been designed to meet the range of power output from 80 to 450 kW (Fig. 6 and 7). The units are of geometrical similar design. They have been extensively tested for endurance both in teststand- as well as in vehicular use. Life expectancy of 8 to 10000 hours is well within reach. The only wear has been found in the shaft rubbing seals. During engine overhawl, these parts can easily be replaced. The V belt for the rotor drive matches the belt life of the generator- and fan belt drives. Trucks have been operated over two million kilometers at satisfactory reliability (Fig. 8).

Comprex supercharging of passenger car engines has been started relatively recent. 1978, an Opel 2,1 liter Diesel engine has first been tested with the Comprex. Excellent response at low speed engine up to a high torque were the convincing features of this first test. In the further development, the Comprex was adapted to meet the higher engine speed range. The somewhat lower supercharging levels, sufficient for passenger car engines, allowed to increase the flow capacity for a given rotor size. Thus, a smaller Comprex matches the demands of the same engine.

As light weight passenger cars need less power (engines of the 1 liter class), smaller units have been added to the existing series. The power range has thus been extended down to 30 kW. This Comprex is equivalent in weight to the turbochargers of equal capacity (Fig. 9 and 10).

A comparison of efficiencies can be seen at Fig. 11. It should be pointed out that there is no efficiency deterioration with down-sizing. The efficiencies given refer to a representative average engine operating point. The compression efficiency based on an isentropic temperature rises as high as 90 %. The combined efficiency reaches 56 % at the best point. One should realize that the wide speed range calls for a sacrifice in peak performance. In the case of optimized performance for narrow range or single point operation, the combined efficiency reaches 74 % which amounts to component efficencies in the middle eighties.

The Comprex for Vehicular Diesel Engines

Present capabilities of the Comprex:

- Engine RPM range for full torque 1: 5
- Engine RPM range between idling and max. power 1:10
- Exhaust temperatures up to 850^{0} C
- No low volume limitation (no Surge)
- Automatic altitude correction
- Instantaneous response under all operating conditions

The Comprex can meet the requirements of the passenger car Diesel engine.

The Comprex offers the greatest advantage in cases where frequent and instant load changes are of significance, combined with a wide engine speed range operation.

Fig. 12 shows the torque curves for different levels of supercharging. In single stage supercharging peak mean, effective pressures up to 20 bars are possible, whereas two stage (combined Comprex turbosupercharger) supercharging yields up to 28 bars.

The largest Comprex units are designed to be used in earth moving equipment. In this application, the capability of producing a high torque, a low engine speed without delay in torque rise is of primary concern. The frontend loader is shown in Fig. 13. Fig. 14 refers to a typical work cycle of loading, transporting and unloading. The best judgment can be made by comparing the performance of the same piece of equipment using the same Diesel engine, the same power shift transmission in one case with the Comprex and in the other with a turbosupercharger. The maximal fuel input for both injection pump is the same. The performance advantage of the Comprex over the turbocharger becomes apparent. The gain in transportation volume was established to be between 9 and 16 %. The difference was due to the ability of the driver. In cases of a more demanding mission, the advantage of the Comprex was even more pronounced. If one considers the cost (equipment amor-

tisation, wages and fuel), the cost difference of the Comprex vs the turbo-charger is totaly insignificant. The drivers expressed their favorable opinion regarding the advantages due to the higher load capability.

Ploughing Tractor

Again the specific power requirement of this type of equipment is well suited for a Comprex supercharged Diesel engine. The Finish Company Valmet has been the first engine and tractor builder to install the Comprex as standard equipment. The production advantage is particularly significant since the thrust force to push the plough varies as much as 50 %, depending on the compactness of the earth. The ability to continue the operation through a segment of increased resistance is of primary importance. The capability of a substantial torque rise with decreasing engine speed represents the essential feature. In case of a stall with turbocharged engines, the tractor has to start up again loosing time and thus production. As fluctuations occur at short intervalls, stalls can become very frequent. Operating at part load at a lower gear ensures the availability of extra thrust to pull through a higher drag section. This means lower utilization of the equipment. More time is needed for ploughing the same area. A power shift gear can overcome some of the disadvantages. However, the added Comprex cost offsets the price of the power shift gear. The Valmet tractor is shown in Fig. 15.

A Steyr Daimler Puch Tractor has been tested with a Comprex utilizing compressed air intercooling. In this case, the torque rise was as much as 38 %. This allowed the tractor to operate at lower engine speed at the normal ploughing speed with a considerable fuel serving. The high torque at low speed engine proved to start up the tractor without excessive clutch abuse.

The Comprex in the Truck Engine

Modern direct injection engines are known for their high efficiency. A further improvement, thus, can only be realized by lowering the engine speed at equal power which means a torque increase by rising the manifold pressure. This again is the domain of the Comprex, capable to lower fuel consumption without additional control means such as a waste gate. Lowering the speed also lowers noise emission. The advantage of fewer gear shiftings and the use of a gear box with fewer stages should be considered in evaluating the economy of the Comprex supercharger.

The Comprex in the Passenger Car

The introduction of Diesel engines in passenger cars has primarely been initiated by the attractive fuel economy. The fact that this engine realizes a stratified charge concept makes it also more favorable with respect to emissions without the need for expensive devices for exhaust gas after treatment. The lower power output at equal displacement and equal weight calls for supercharging. The torque shape and response should be as good as with the standard gasoline engine. The Comprex meets this requirement. The passenger car engine has its most frequent use in the middle range of rpm at about one third of its peak torque. At this regime, the thermal efficiency of the gasoline engine is about two thirds of the peak efficiency. The Comprex supercharged Diesel engine, both with indirect and direct injection at equivalent part load, has a thermal efficiency equal or better than the gasoline engine at peak torque.

The fuel savings with Diesel powered passenger cars are therefore substantial. The mixture enrichment in the gasoline engine warmup period makes up for a 20 % fuel consumption increase in the first 5 km. The Diesel engine does not need such modifications.

In 1978, the first Comprex supercharger fitted to a passenger car engine had the Comet combustion chamber. The torque curve 1, Fig. 16 shows the moderate gain reached at that time which agreed with the torque curve of the turbocharged engine. In driving the car, the response made the car seem considerably more powerful.

Based on the experience with Comprex supercharged truck engines, the bmep compared to naturally aspirated engines was still quite moderate. Due to the larger engine speed range of passenger cars, compared to the relatively narrow speed range of truck engines, made the torque at low engine speed fall off below acceptable levels. High torques at one quarter of the max. engine speed are taken for granted in gasoline engines.In this range additional development efforts were concentrated. It was found that strong pressure pulsations occurred in the engine exhaust gas collector which disturbed the Comprex performance. A modification of the stator geometry has been established to correct this disadvantage. Curve 2 in Fig. 16 indicates the substantial gain. At the same time, the flow capacity has been increased, thus allowing to reduce the rotor diameter from a diameter of 112 mm to 93 mm. A Comprex weight reduction to about 60 % of the previous weight, needless to say, brings several advantages.

The intake manifold pressure ratio in the low speed range is now between 1.4 and 1.5. If a charge air cooler is incorporated, a density ratio of about 1.4 means a bmep of 10 bar at .25 of peak engine speed. In the mean engine speed range a bmep of nearly the double of its naturally aspired counterpart was obtained. At max. engine speed, the bmep gain is falling to about 30 to 40 %. Here the cycle peak pressure represents the limiting factor. One should be aware that engine size and gear box ratios should be selected carefully. In doing this, the criterion of available excess power at a car speed of 40 km/h to produce an acceleration of 0,7 m/sec^2 should be considered. This would permit a reduction of the engine displacement in the Opel engine to 1,64 liter or to increase the vehicle mass proportionally. The fuel consumption has been measured on the vehicle dynamometer with a 2 liter engine assuming the heavy vehicle. The expected fuel consumption for the Opel with the 1,64 displacement engine has then been established by calculation. The results are presented in Fig. 17 as a comparison.

In order to fully benefit from the potential fuel saving with no sacrifice in car performance, the following requirements will have to be met:

- A high degree of supercharging in the entire engine operating speed range.
- Availability of full torque within 0.5 sec.
- High thermal efficiency in the medium engine speed range at a bmep level of 4 bar to be within 10 % to the peak efficiency of 32.6 % (equal to 260 gr/kWh). Future direct engines will be 15 % better.

High torque at low engine speed allows engine utilization with very much lower noise emissions.

The Comprex has received much attention lately for its potential use in passenger car Diesel engines. The 3 liter 6 cylinder engine in the Daimler-Benz as well as the 1.2 liter 3 cylinder engine in the VW represents the extremes of the wide range of application evaluation. Both engines with indirect and direct injection, with and without charge air cooling are being tested. There are cars with manual gear shift and with automatic transmission. Engines have to be designed sufficiently rugged to be supercharged to a manifold pressure ratio of 2.Air cleaner and exhaust mufflers have to be laid out for low pressure drop at the larger air and exhaust gas flows.

An application of a small Diesel engine supercharged by the Comprex is being tested in the Steyr-Daimler-Puch jeep. This vehicle, shown in Fig. 18, demonstrates the specific advantages of the new system having a high torque in the full range of engine speed.

Fig. 19 represents a typical Comprex installation on a vehicular Diesel engine. In order to simplify the beltdrive, the Comprex is placed in such a way that the belt pulley is in the plane of the V beltdrive for the electric generator. The Comprex is bolted to the exhaust gas collector. This eliminates the need for flexible expansion joints. As the drive power is quite small, the belt force driving the Comprex rotor is insignificant. An idler pulley is used to adjust the belt tension. The piping of the compressed air is flexibly mounted with rubber sleeves. For starting the engine, a simple small alternate air intake valve is provided in the compressed air duct. The spring loaded valve opens automatically by the vacuum when the engine is turned over by the electric starter. The over all installation is simple since there is no need for a hot bypass valve with controls. Also no control means are necessary to limit the fuel input during transients.

Summary

The Comprex represents a new principle suitable to supercharge vehicular Diesel engines with requirements for a wide speed range operation and for rapid response to load changes. The device operating at low speed is simple in its design. Emission- and the noise standards can be met.

Fig.1. Comprex Rotor Casting Facility.

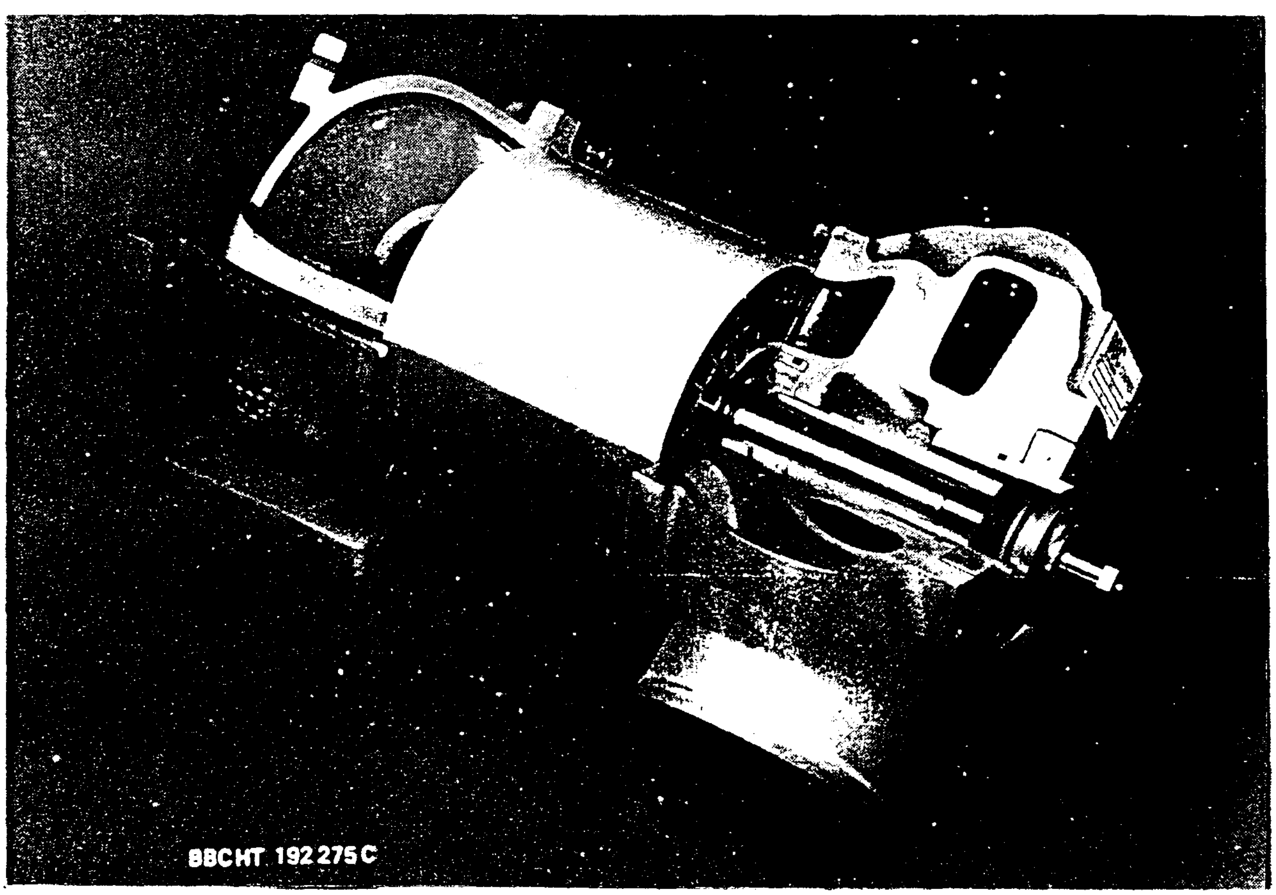

Fig.2. Comprex Supercharger.

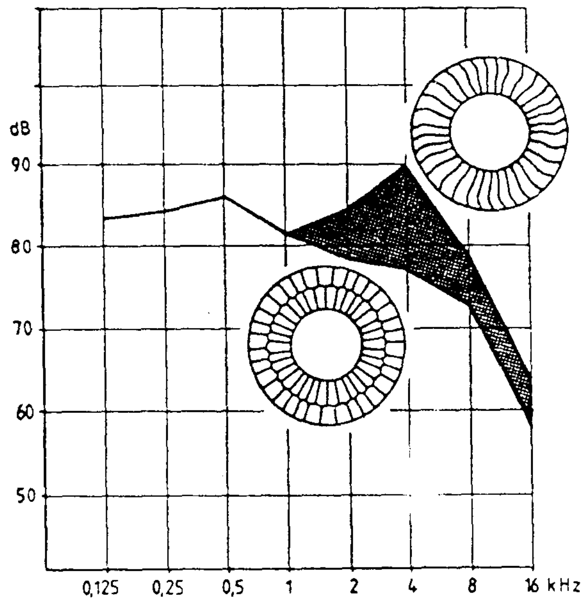

Fig.3. Noise Emission Spectra for different Rotor design Configurations.

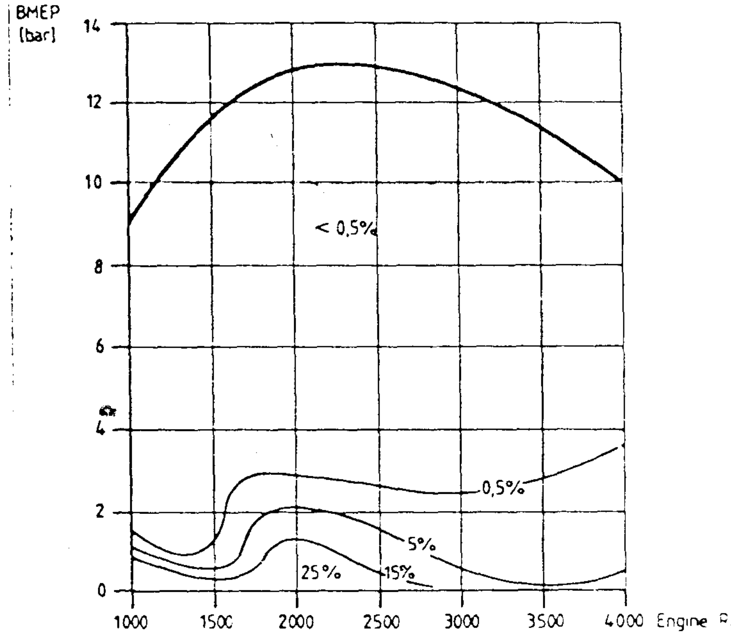

Fig.4. Diesel Engine with Comprex Engine Performance Map with internal Exhaust Gas Recirculation.

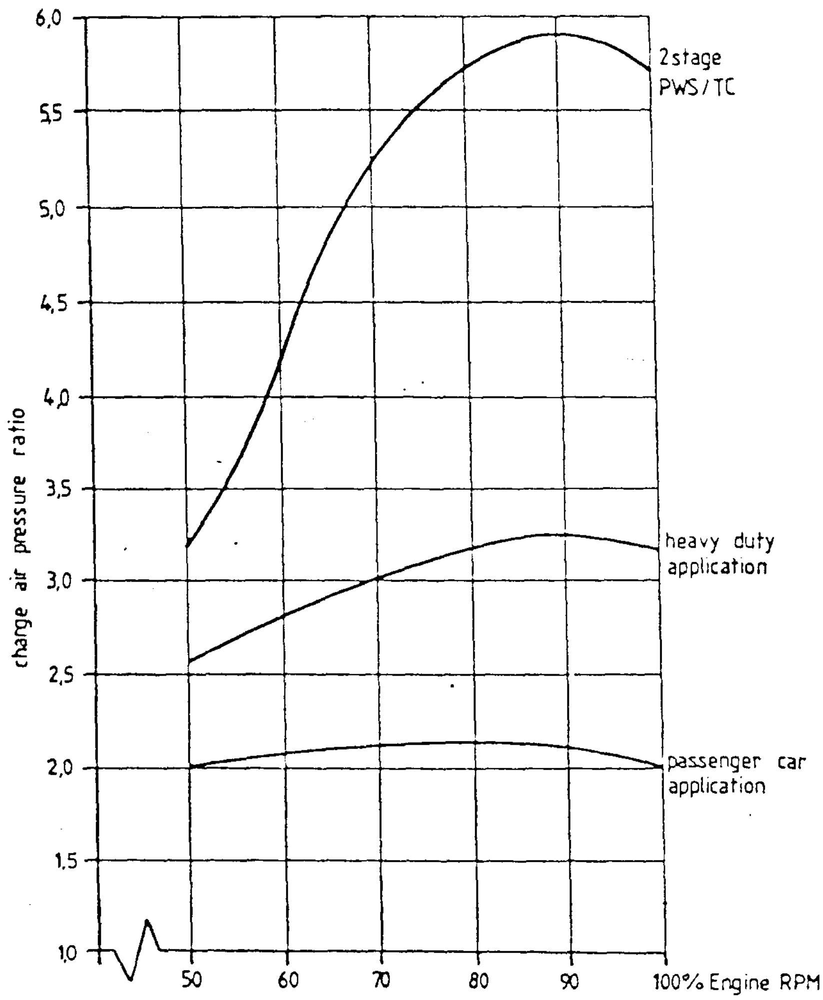

Fig.5. Supercharging Pressure.Ratio vs Engine Speed with Comprex Supercharging.

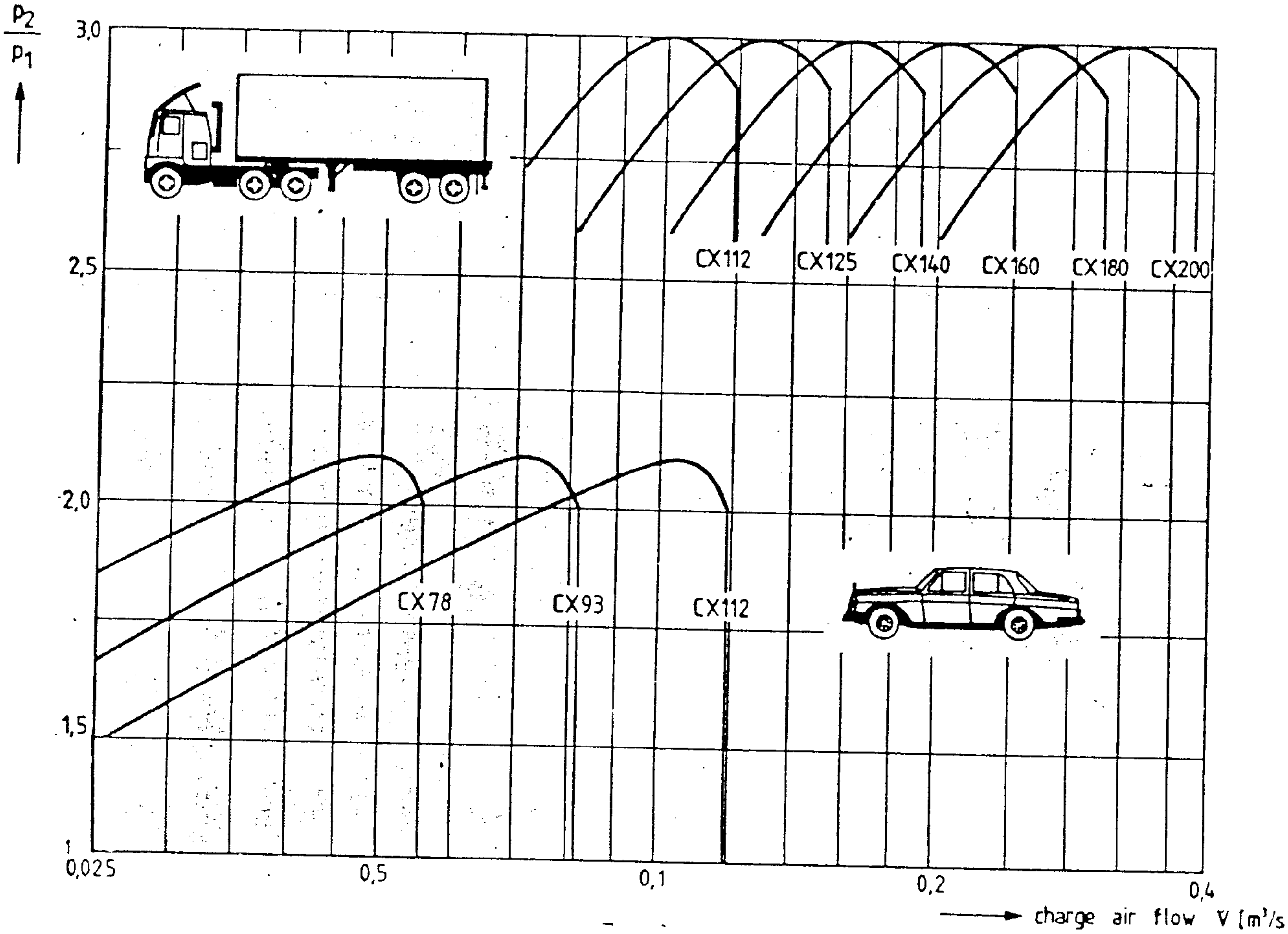

Fig.6. Volume Flow Capacity Ranges of Different Comprex sizes for Truck- and Passenger Car Diesel Engines.

Fig.7. Comprex Supercharger Models.

Fig.8. Saurer Truck with Comprex Supercharger.

Fig.9. Small capacity Comprex for Passenger Cars Diesel Engine.

Commercial vehicle range	Nominal engine output [KW]	Weight [kg]
CX 200	250 - 450	40
CX 180	200 - 350	32
CX 160	160 - 280	24
CX 140	125 - 225	18
CX 125	100 - 180	14
CX 112	80 - 140	12
Car range		
CX 112	60 - 90	10,5
CX 93	42 - 62	7,0
CX 78	30 - 45	5,7

Fig.10. List of Comprex Model Sizes
with Engine Power Rating.

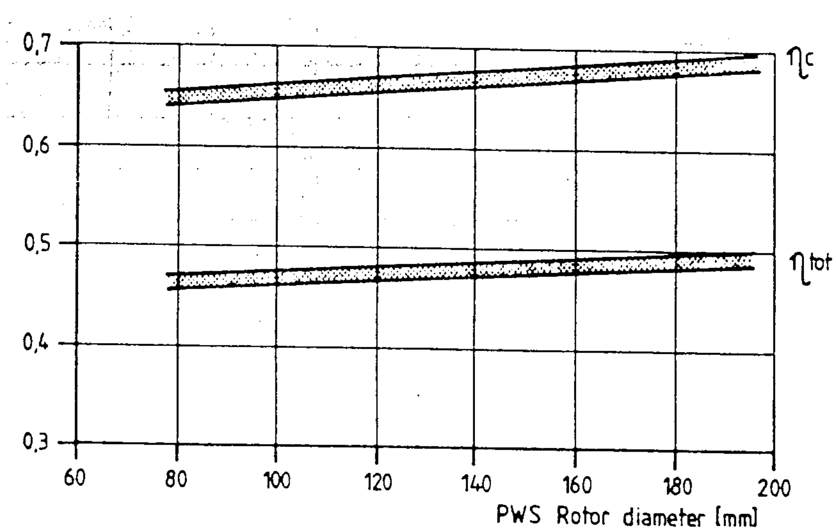

Fig.11. Equivalent Comprex Efficiency
vs. Comprex Size
Compression Efficiency
Overall Efficiency.

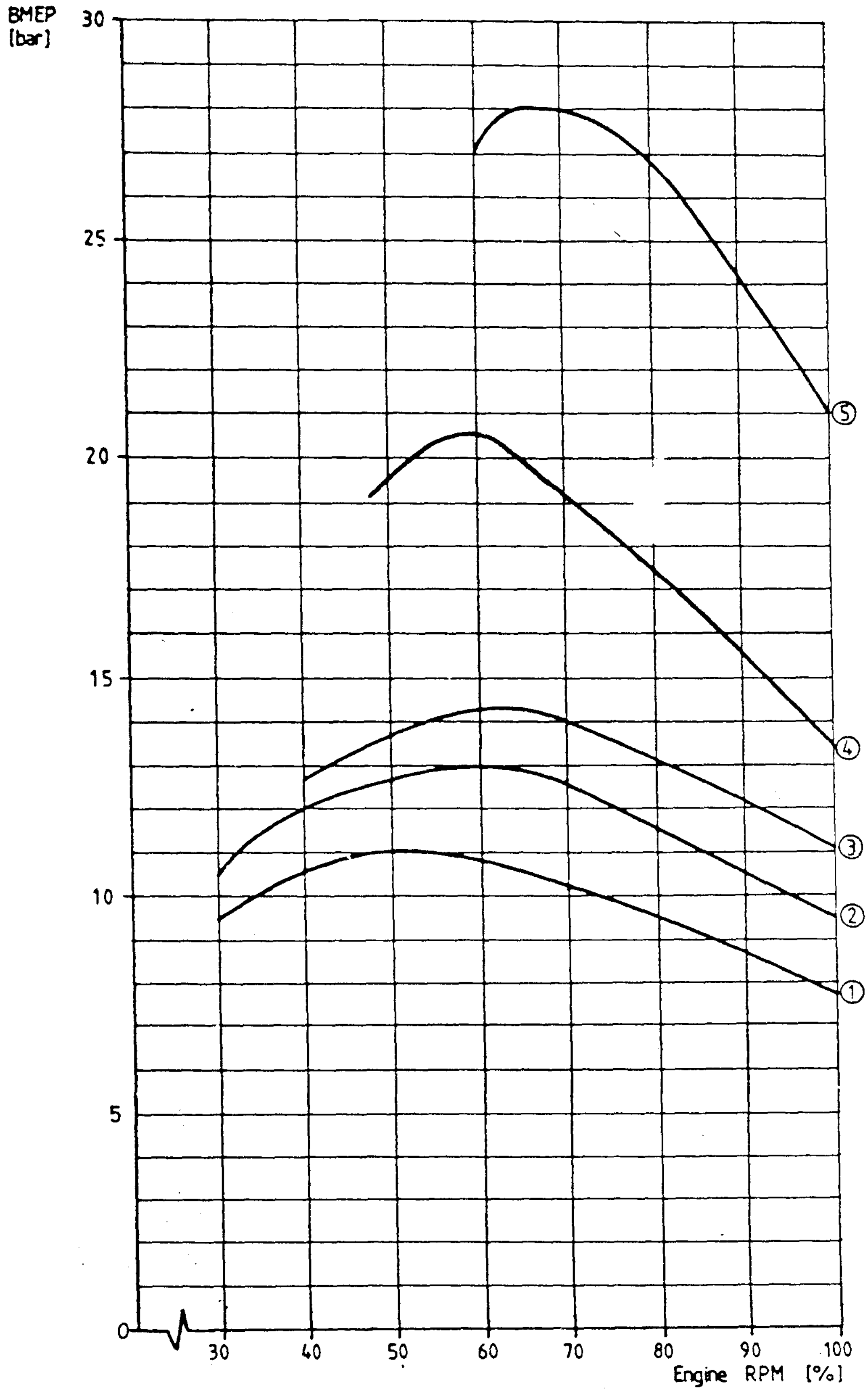

Fig.12. Mean effective Pressures vs. Engine speed for different levels of Comprex Supercharging.

Identification of curves:
Intercooling is applied in all cases except Curve 1.

1) Passenger Car
2) Passenger Car
3) Truck Engine
4) Heavy Duty Truck Engine
5) Heavy Duty Truck Engine with Comprex and Turbocharger.

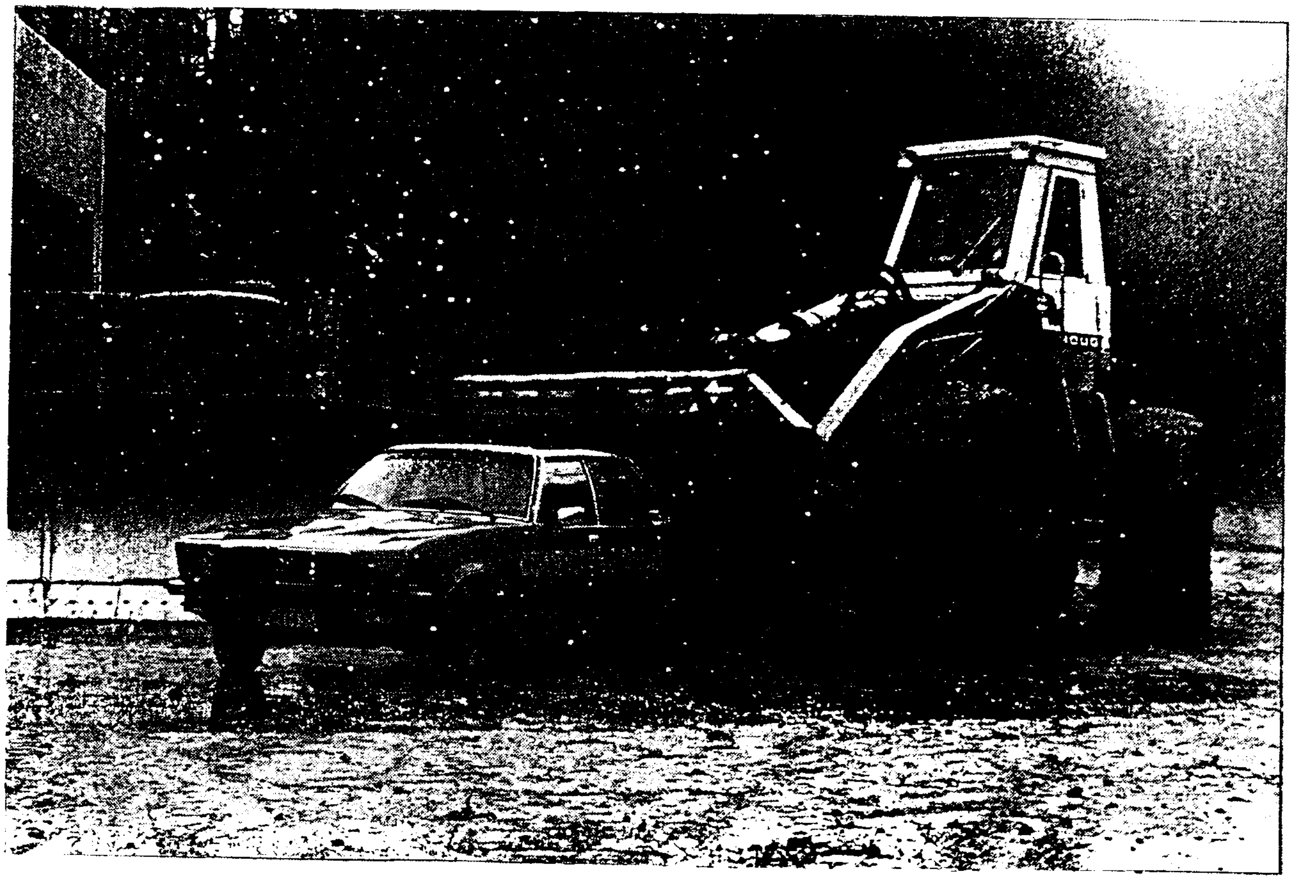

Fig.13. Frontend Loader.

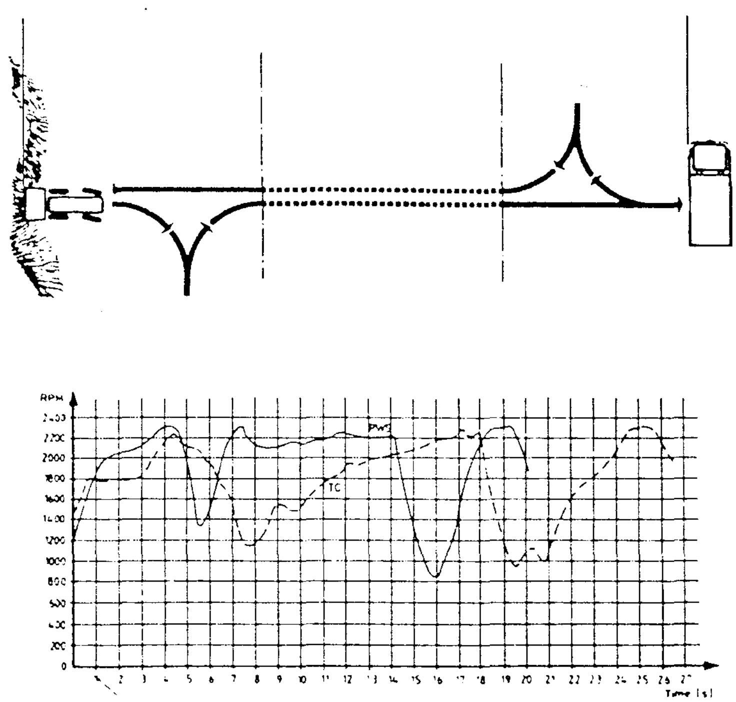

Fig.14. Frontend Loader Mission.

Fig.15. Valmet Tractor.

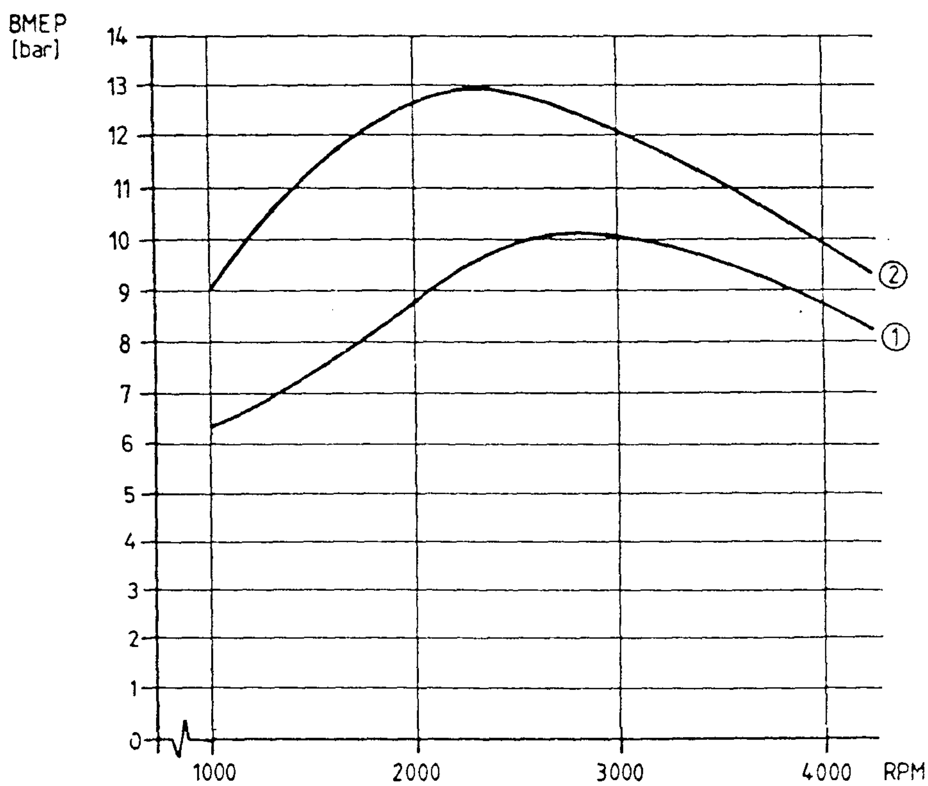

Fig.16. Comparison of BMEP Curves vs. Engine speed
1) First Test of Comprex on Opel IDI Engine
2) Improved Matching of Comprex on Opel IDI Engine.

Test Vehicle : Opel Record

Inertia weight : 3000 lb

Fuel economy	2,3 l	1,64 l
Urban Test	30,0	36,8 mpg
Highway Test	38,8	44 mpg
CFE	33,4	39,7 mpg

Emissions		
HC	0,26	[illegible] 1 gpm
NO_x	0,64	[illegible]9 gpm
CO	1,5	[illegible] gpm
Particulates	0,32	0,48 gpm

Drive-by Noise	
(ISO 362)	74 dBA

Fig.17. Opel Record Passenger Car Performance and US Emission Test Comparison for equal Car weigt and different engine sizes.

Fig.18. Steyr - Daimler - Puch 4-wheel - Drive Experimental Car.

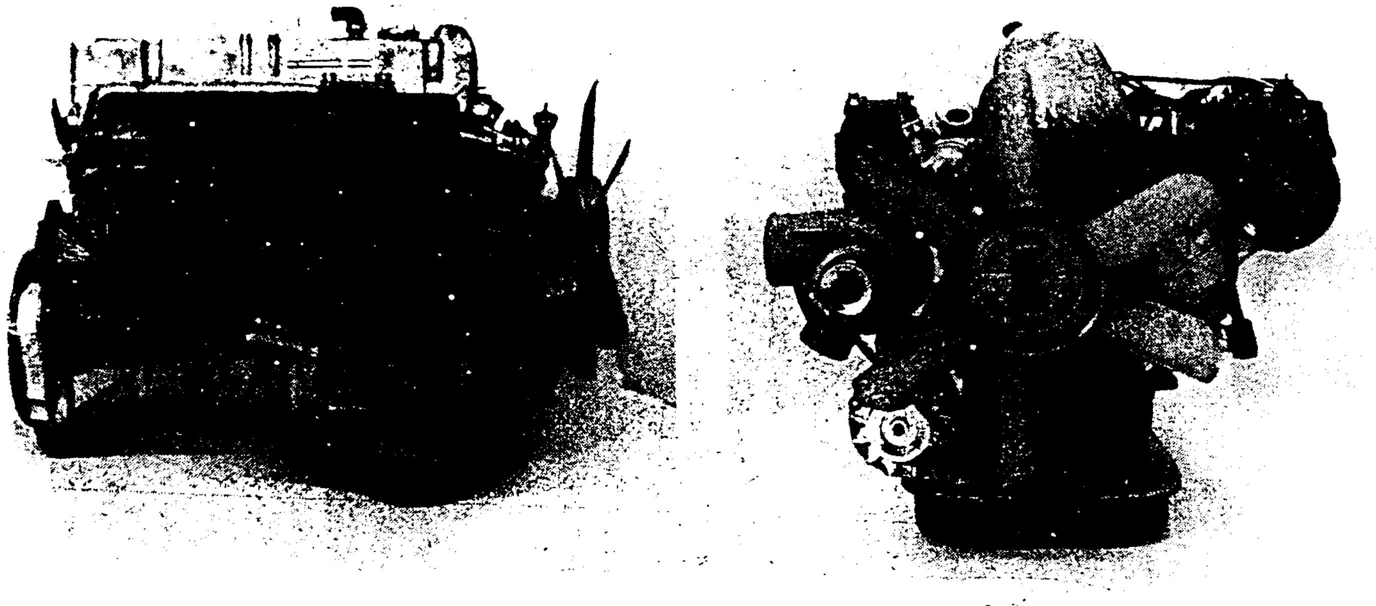

Fig.19. Comprex Installation on vehicular Diesel Engine.

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Fig. 1.

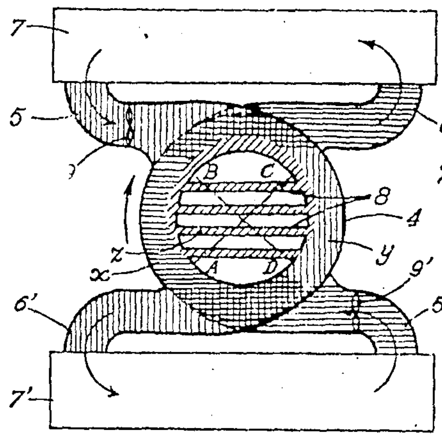

Fig. 2.

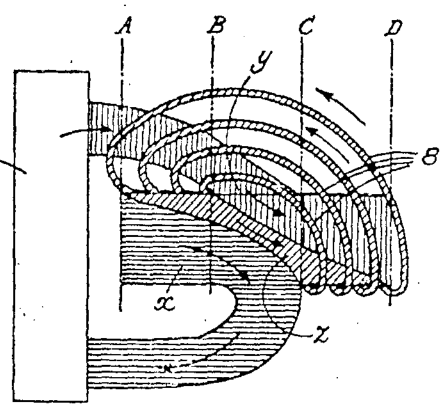

Fig. 3.

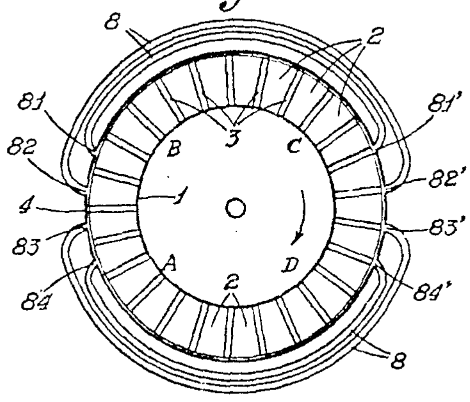

Fig. 5.

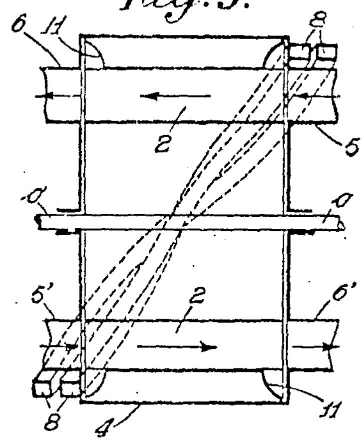

Fig. 4.

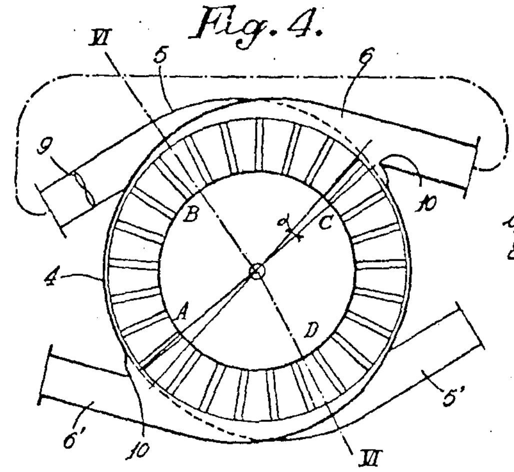

Fig. 6.

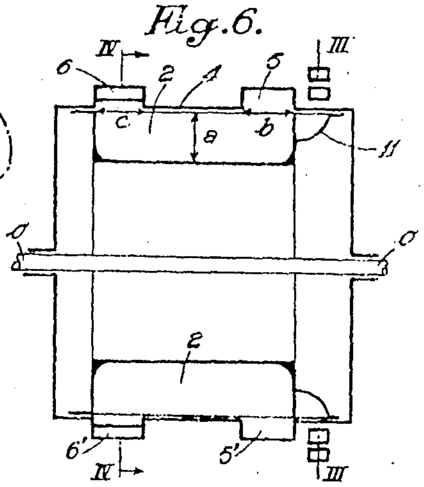

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Fig. 7.

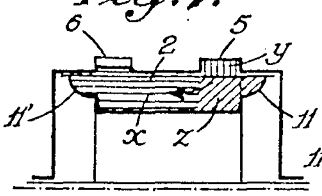

Fig. 8.

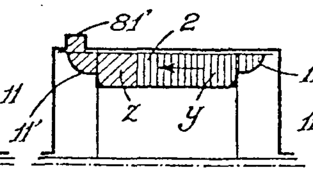

Fig. 9.

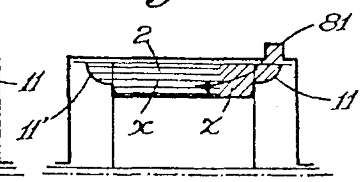

Fig. 15.

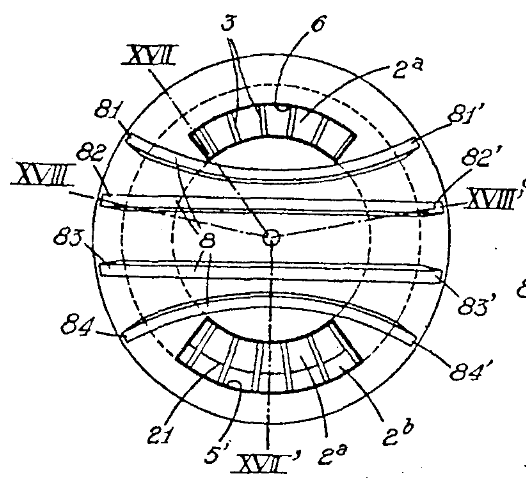

Fig. 17.

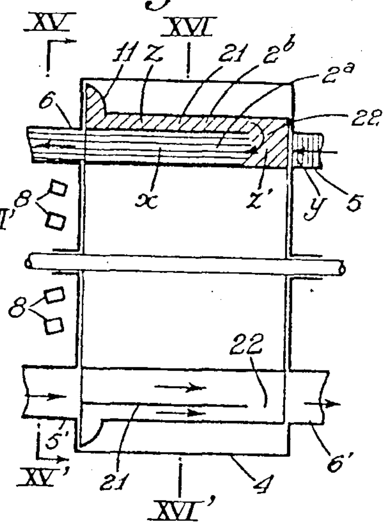

Fig. 16.

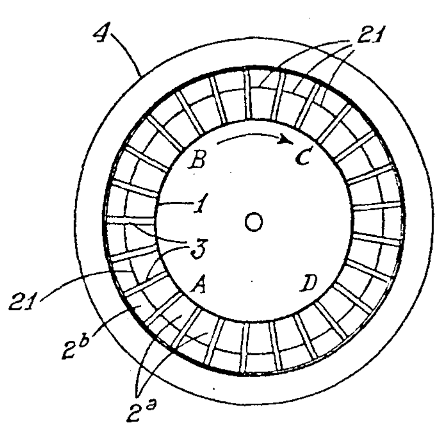

Fig. 18.

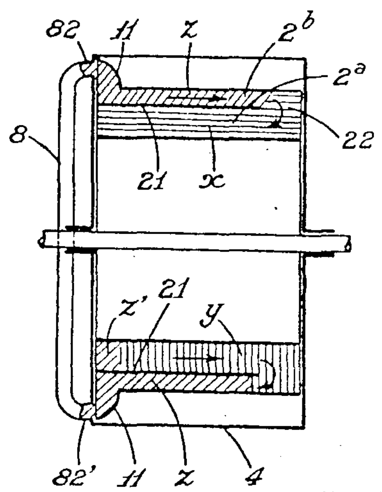

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Fig. 10.

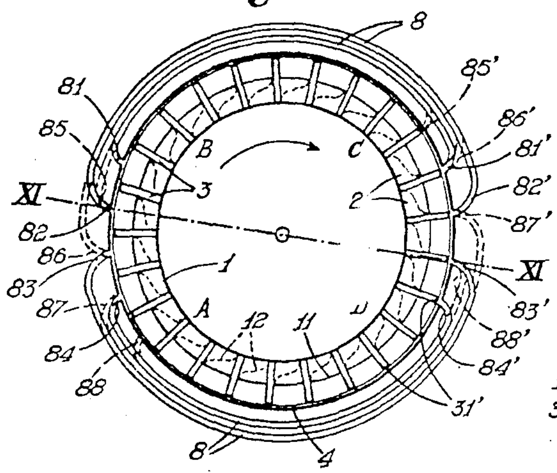


Fig. 11.

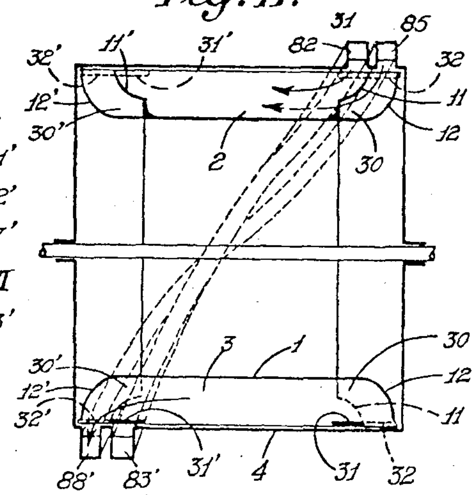


Fig. 12.

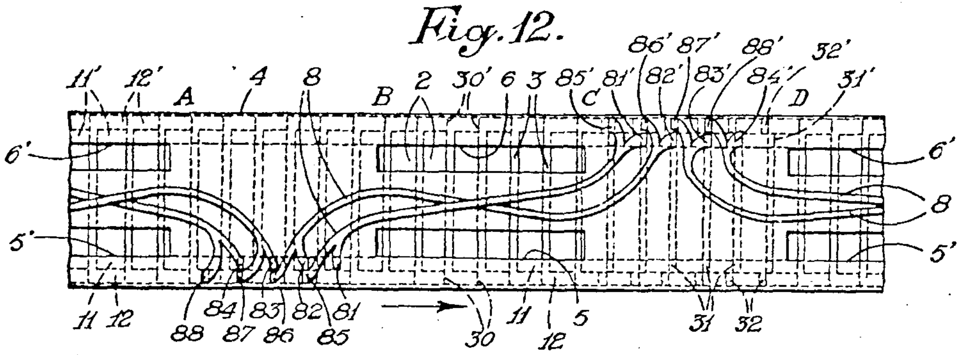


Fig. 13.

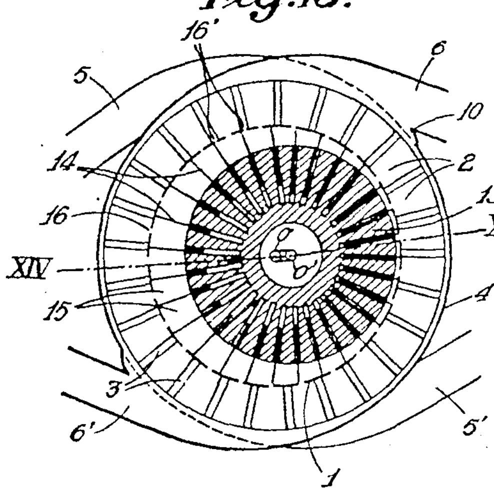
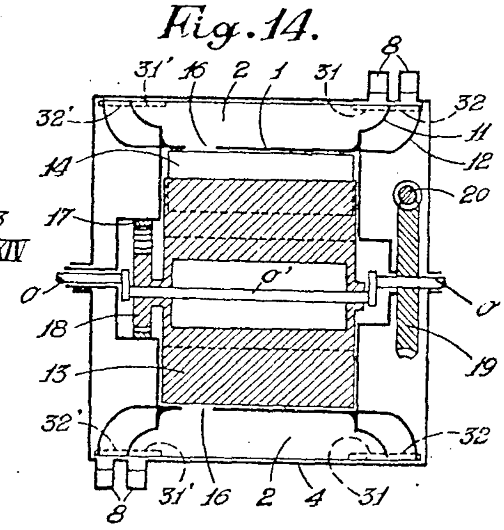


Fig. 14.



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UNITED STATES PATENT OFFICE

2,045,152

PROCESS OF AND APPARATUS FOR PERFORMING CONVERSIONS OF MECHANICAL AND THERMAL ENERGY

Albert François Lebre, Paris, France

Application March 16, 1934, Serial No. 715,875
In France March 27, 1933

32 Claims. (Cl. 62—170)

The present invention relates to processes and apparatus wherein use is made of gases, in cycles comprising compression, heat exchange at a constant and relatively high pressure, expansion, and a heat exchange at a relatively low pressure, for the purpose of producing heat or cold, for example.

Since a gas absorbs energy when being compressed and releases energy when expanded, it is essential in thermodynamic cycles that the energy contained in the expanding gas be recovered as fully as possible, in order that it may be utilized again for assisting compression. The processes and apparatus proposed hitherto for this purpose have not given wholly satisfactory results, owing to important losses of energy arising during the stages of compression and expansion.

In one class of process, use was made of independent mechanical means, to produce compression and expansion separately. This class of process involved a complete separation of the currents of gases, respectively undergoing compression and expansion, by means of members such as pistons reciprocating in cylinders. The losses of energy entailed by such members, more particularly owing to friction, were too great owing to the additive effect thereof relatively to the total work of compression and expansion.

It is desirable to apply the action of the expanding gas upon the gas to be compressed whilst interposing therebetween the smallest possible number of moving parts. In this connection it has been proposed to omit all intervening moving parts, and to transmit pressure from the expanding gas directly to the gas to be compressed. Hence, in another class of process it was proposed to expand the gas by partial withdrawals therefrom, at decreasing pressures, whilst said gas was contained in a closed chamber and conversely to compress the gas contained in another closed chamber by successive additions of gas, at progressively increasing pressures which were always higher than those obtaining in the receiving chamber, this being effected by connecting through suitable ducts, chambers containing the gas to be expanded with chambers containing the gas to be compressed. These ducts, between which and the said chambers a relative motion was provided, acted as distributors progressively to lower the pressure of the gas to be expanded, said gas being contained in a series of chambers, and gradually to increase the pressure of the gas to be compressed in another series of chambers, thus ensuring a rational pressure exchange. The subsequent opening of the chambers into the high pressure zone allowed the compressed gas to issue from the chambers and to be displaced into a constant pressure heat exchanger. This displacement, brought about by the gas issuing from the heat exchanger, only called for an expenditure of energy equal to that due to the pressure losses in the circuit and could be produced by a fan.

It was thus sought to make use of the energy released by expansion to compress an equal volume of gas to a like pressure, in order to limit the mechanical energy used to that required to compensate variations in gas volume due to heat exchanges and losses due, more particularly, to leakages.

If, in this latter class of process, losses by friction due to moving partitions were eliminated, on the other hand, other losses were created which involved practical failure due to the intermingling of gases to be compressed and gases to be expanded, and to unavoidable leakages more particularly such as arose, from the partitioning of the gas currents in the chambers, along the whole of the surfaces separating the chambers from one another.

The object of the process according to my invention is to overcome the disadvantages innate in these two classes of processes, and to combine the advantages derived from partitioning with those resulting from the direct action of the gas currents upon each other.

The invention is based upon observation of the fact, that an ideal means of separating the principal gas currents (i. e. the currents of gas undergoing compression and of gas undergoing expansion) is provided by the gas current, resulting from pressure exchange between the expanding gas and the gas to be compressed, provided it be found possible to prevent said gas current from mixing and intermingling with the principal currents, during its passage outside the connecting ducts.

To this end the process according to my invention consists in so maintaining and guiding the pressure exchange current, that it flows in the same direction as the two principal currents and that, while remaining distinct therefrom, it is interposed in the fashion of a gaseous partition, between said two currents which are at different temperatures and flow respectively towards and away from the heat exchanger. When the process is performed in a rotary apparatus comprising a ring-shaped series of chambers moving relatively to a fixed distributor or a sta-

tionary series of chambers with a moving distributor, I am able so to regulate the speed, the orientation and the position of the pressure exchange current that said current flows through the rotary apparatus, between the stages of compression and expansion, without leaving said apparatus and hence without participating in the heat exchange.

In order that the pressure-exchange current may act after the fashion of a gaseous partition it is necessary that, in contrast to previous practice with processes of the second class aforesaid, the current of gas from the expansion phase to the compression phase be kept as continuous as possible, in regard to speed, orientation and position. More particularly the connecting ducts must at all times communicate with one at least of the two phases and their orifices must be so arranged, relatively to the chambers, as to obviate any short-circuit, or leakage, between two neighbouring chambers, during the phases of expansion or of compression.

To this end the orifices of the connecting ducts are given a width equal to that of the partitions separating the inlet ports of the chambers so as to allow of said orifices being closed, during their passage from one chamber to the next, thus averting a short-circuit. Furthermore the arrangement is such that when one end of a duct is closed by a partition, the opposite end of said duct is situated centrally of a chamber. Stoppages and jars liable to promote the mixing or mingling of the fluids are thus avoided, and the continuity of the variations of pressure is enhanced.

The pressure-exchange current thus delivered in a practically constant manner, through the connecting ducts, is given a substantially constant direction, relatively to the currents in the expansion and compression phases, owing to suitable orientation of the orifices of the ducts which communicate with the chambers containing the main currents. Said orifices are distributed exclusively over that portion of the casing, which closely surrounds the chambers, in order to avoid that the ducts should communicate periodically with the admission and exhaust pipes.

Since the circuit of the pressure-exchange current is completed through the chambers into which the compressed and heated gas is displaced by the cooled compressed gas to be expanded, before said compressed and heated gas escapes into the heat exchanger, the openings of the admission and exhaust pipes are shaped and arranged to ensure the continuity of the pressure-exchange current during this phase, and to reduce the speed at which the principal currents are admitted and exhausted, in order to avert too abrupt a passage from one phase to the next succeeding one.

The gaseous current formed between the expanding gas and the compressed gas by said pressure-exchange current may be so extended as to complement the material partition separating adjacent chambers and to prevent leakage between them along the periphery of the casing. To this end, the vanes or partitions of the rotor separating the successive chambers are made hollow, and the spaces thus formed within these vanes or partitions are caused to communicate with connecting ducts, wherein a pressure obtains, which differs slightly from the pressures within the chambers to be separated. On the expansion side this pressure will be slightly lower than those obtaining on either side of the partition under consideration, and on the compression side it will be slightly higher, in order to by-pass a portion of the pressure-exchange current through the partitions and connecting ducts, without impairing the continunity of said current, and thus to complement the material partitions afforded by the partitions or vanes, by means of gaseous currents located along the edges of said vanes. To the gas currents at different temperatures which are to be separated, and to the pressure-exchange current which is to ensure their separation, must be added, within the apparatus, a current required to compensate the variations of volume, due to changes of temperature, undergone in the heat exchangers situated outside the apparatus; i. e. between the moment when a gas current issues from the apparatus and the moment it re-enters said apparatus after having participated in a heat exchange. According to the invention, use is made of such compensating current, to enhance the continuity of the pressure-exchange current. As is the case with the latter, said compensating current flows parallel to the principal current, a result advantageously obtained by associating with the chambers of constant volume chambers of varying volume adapted to be added thereto or substracted therefrom, as need may arise.

According to one practical and simple embodiment of the invention the chambers arranged in ring formation and mounted for rotation around the axis of the ring, are closed by a casing during the stages of compression and expansion, said casing merging into volutes in two diametrically opposed zones to guide the principal currents out of the apparatus during the stages of displacement at constant pressure. The zones of compression on one hand, and of expansion on the other hand, communicate with each other through connecting ducts secured to the casing. The vanes or partitions separating the chambers are hollow and the pressure exchange current is by-passed through them whilst they traverse the zones aforesaid.

A compressor rotor having blades thereon is mounted eccentrically within the ring, the variable capacity required to create the compensating current flowing parallel to the gas current undergoing compression, being constituted by the space comprised between successive blades.

My invention will now be described in greater detail with reference to the accompanying drawings, which illustrate diagrammatically and by way of example the principle of the invention, some preferred embodiments thereof and explanatory diagrams.

Fig. 1 is a diagrammatic view of the apparatus as a whole.

Fig. 2 is a diagram illustrating the functions of the pressure-exchange current.

Figs. 3 and 4 are diagrammatic sections taken through planes at right angles to the axis of revolution of a rotary apparatus, said sections being taken respectively on lines III—III and IV—IV of Fig. 6.

Fig. 5 is a diametrical section showing a modification.

Fig. 6 is a diagrammatic section on line VI—VI of Fig. 4.

Figs. 7, 8 and 9 are explanatory sectional views similar to Fig. 6.

Fig. 10 is a sectional view at right angles to the axis of rotation of an apparatus provided with means for by-passing a portion of the pres-

sure-exchange current through hollow partitions or vanes.

Fig. 11 is a section on line XI—XI of Fig. 10.

Fig. 12 is a development of the periphery of the casing and of the connecting ducts of said apparatus according to Figure 10.

Fig. 13 is a section similar to Fig. 4, illustrating the arrangement of a compressor designed to provide a compensating current adapted to cooperate with the pressure-exchange current.

Fig. 14 is a section on line XIV—XIV of Fig. 13.

Fig. 15 is a sectional elevation on line XV—XV' of Fig. 17, illustrating another form of apparatus.

Fig. 16 is a section on line XVI—XVI' of Fig. 17.

Fig. 17 is a radial cross-section on line XVII—XVII' of Fig. 15 and

Fig. 18 is a radial cross-section on line XVIII—XVIII' of Fig. 15.

In the drawings 1 indicates the rotor comprising a ring of chambers 2 separated from each other by partitions or vanes 3, said rotor revolving in a casing 4 provided with inlet pipes 5, 5' and outlet pipes 6, 6' connected with a high pressure heat exchanger 7 and a low pressure heat exchanger 7' respectively.

The rotor revolving as indicated by the arrow (Fig. 1), it is assumed that compression takes place from A to B, and displacement under high pressure from B to C, the compressed and heated gas being cooled under constant pressure in the heat exchanger 7, and led back to the rotor between B and C, wherein it expands from C to D. Upon having resumed its original pressure the cooled gas is delivered between D and A, into the exchanger 7' wherein it becomes heated under constant pressure; thence it is returned to the rotor, recompressed from A to B and started upon a fresh cycle. In the case of a chamber performing a complete revolution the successive stages or phases are therefore: compression (A—B), displacement (B—C), expansion (C—D) and displacement (D—A).

The ducts interconnecting the phases of expansion C—D and of compression A—B are shown at 8. Each of them is so arranged as temporarily to connect a chamber under expansion with a chamber under compression at a lower pressure. A transfer of fluid therefore takes place through said connecting ducts, from the chambers under expansion to the chambers under compression with which they are connected in succession. Whilst following the arc A—B, a chamber containing the fluid to be compressed will therefore receive a slight addition of fluid from each connecting duct, and will undergo a series of partial compressions adapted to raise the pressure of the fluid contained therein to the final pressure required. Likewise, whilst travelling along the arc C—D, some of the fluid to be expanded will issue into each duct 8, until such fluid has resumed its initial pressure.

Since the fluids in A—B and C—D are at different temperatures, it is essential that they do not mix, either in the chambers 2 or in the displacement zone B—C where admission and exhaust take place simultaneously under constant pressure. According to my invention, I attain this result by maintaining in the ducts 8 and in the chambers 2 a continuous current adapted to separate the two main currents, to accompany them through phases A—B, B—C and C—D and to act somewhat like a gaseous screen or partition between them, without however entering the heat exchanger 7.

The part played by such a gaseous partition will become apparent from a consideration of Figs. 1 and 2 wherein the horizontal shading lines indicate the course followed by the fluid x to be compressed, and the vertical lines the course of the fluid y to be expanded, the oblique lines indicating the pressure-exchange current z. As is illustrated most clearly in the diagram of Fig. 2, wherein the development of arcs A—B, B—C, C—D is shown as abscissae, and the width of the ring of chambers as ordinates, the arrangement is such that current z shall flow in the ducts 8,—urge the fluid to be compressed into the chambers comprised in the zone A—B,—separate in zone B—C the outgoing, compressed and hot fluid from the incoming fluid, similarly compressed but cooled,—and flow back thereafter into the ducts 8 under a pressure equal to that of the gas to be expanded in the zone C—D.

The gaseous current may flow through the ring of chambers, from the centre towards the periphery thereof or, inversely, from the periphery towards the centre, or again in a direction parallel to the axis of rotation. This latter arrangement, whereof two alternative constructional embodiments are shown in Figs. 5 and 6 affords certain advantages as regards simplicity of construction, but it is to be understood that use of the present invention is by no means limited thereto.

In the case of Figs. 5 and 6 the circulation of the gaseous streams through the heat exchangers, the inlet and outlet pipes and the rotary apparatus, is ensured by suitably located fans 9, 9' (Fig. 1). The openings of the inlet and outlet pipes 5, 5' and 6, 6' extend throughout the length of arcs B—C and D—A. In Fig. 5 said pipes are arranged to extend over the parts B—C, D—A of the ring of chambers in a direction parallel to the axis of rotation, whilst in Fig. 6 they enclose the peripheral edges of said parts, and are voluted as shown in Fig. 4.

The connecting ducts 8 are external to the rotor. They likewise open into the casing 4 and are arranged on one side or on both sides of the same.

The ends of the chambers 2 are provided with distributing ports 11, 11', in the shape of spoons for example, which on passing before the orifices of the ducts 8, connect the orifices 81, 82, 83, 84 (Fig. 3) at one end thereof with the chambers comprised in zone A—B, and orifices 81', 82', 83', 84' at the opposite end of said ducts with the chambers comprised in zone C—D.

In order that the flow of the gas currents shall be more uniform the orifices 81—84, 81'—84' of ducts 8 are given the same width as the vanes or partitions 3 (Fig. 3), thus averting the short-circuits which would necessarily arise if the ducts were of greater cross-section. The occlusion caused by the passage of the vanes over the orifices of the ducts 8 only lasts a fraction of a second and the effect thereof may be lessened if care is taken that the moment one end of a duct 8 is closed by a vane 3, the opposite end is at the center of a chamber. In this way both ends of the ducts can never be closed simultaneously, whereby the continuity of operation is maintained and the progressiveness thereof is doubled. Indeed, whilst a chamber under expansion (arc C—D) communicates with one particular duct, the other end of said duct discharges successively into two consecutive chambers comprised in A—B, whereby two partial compressions are obtained during the pressure drop occurring in the corresponding arc. With

this arrangement likewise two chambers under expansion will deliver successively into one same duct, whilst it is connected with one chamber under compression. As a result, the number of partial expansions and of partial compressions is twice that of the number of ducts in use, i. e. eight in the case of the example shown in the drawings, losses of energy and speed of circulation thus being reduced.

As is shown in the drawings, the orifices of the ducts 8 only extend over the parts of arcs A—B and C—D which are closely surrounded by the casing 4, thus preventing said ducts from communicating with chambers not yet closed by the casing, and making it possible moreover, to provide delayed admission and exhaust.

To secure the progressive flow of fluids from one stage to the next one without any jar, means are provided to ensure that, over the whole length of arcs B—C and D—A, the velocity of the gas entering the chambers 2 on one hand, and its outgoing velocity from said chambers on the other hand, shall be equal in magnitude and orientation to the speed assumed by the gas within the chambers under the combined influence of the movement of said gas in said chambers and of the latter's rotation. To ensure this result the inlet pipes 5, 5′ and the outlet pipes 6, 6′ are arranged in tangential relationship to the ring, and their cross-section is varied in accordance with a parabolic function.

The gas admitted into a chamber at the end of a phase of displacement B—C or D—A is endowed with a certain velocity. The casing must however close this chamber in order to initiate the next phase (expansion or compression). In order to avert jars, the internal edge of the exhaust pipe has been given the form of a nose 10 (Fig. 4), whose point is slightly spaced from the ring of chambers and is directed as the component of the velocities of the escaping current, the internal face of said nose gradually merging into a tangent to the ring of chambers.

Instead of using a nose such as 10, I may also retard the velocity of the gas before closure, by effecting such closure by means of a member comprising parallel blades adapted to cause a pressure drop.

Likewise, in order to retard the velocity at the end of the phase of displacement it is desirable that inlet pipe 5 should close before exhaust pipe 6. This off-setting, indicated by angle α on Fig. 4, is usually of a value approximating to that of the angle covered by the nose 10.

The chambers themselves are shaped so that their cross section a, taken at right angles to the direction of gas flow, is approximately equal to their admission area b and to their exhaust area c (Fig. 6). In this figure, admission and exhaust take place across the periphery of the ring of chambers. Alternatively they may of course be effected through the sides or through the interior of the ring.

In operation the current hereinbefore called the pressure exchange current will be set up alongside of and parallel to the principal currents for a great part of their course, owing to the continuity of circulation of the gas currents and to the means provided for guiding the same. If a chamber 2 (Fig. 7) be considered at the moment when the compression phase has just ended, the horizontally shaded zone indicates hot compressed gas as it begins to escape through 6, the vertically shaded zone denotes the compressed but cooled gas admitted through 5, and the oblique shading lines indicate the pressure-exchange current z, which has been admitted into the chamber by the ducts 8 during the compression phase A—B, and has compressed the gas contained therein. During the displacement phase B—C, the current z travels from one end of the chamber to the other whilst remaining interposed between the warm, outgoing current and the cold, incoming current. Velocity is so regulated, that pipe 6 is closed when current z, reaches the opposite end of the chamber (Fig. 8), whereupon orifices 81′, 82′, 83′, 84′ of ducts 8 are brought in succession opposite the port 11′, and each of them allows a portion of the current z to pass therethrough. Since the opposite ends of ducts 8 open into chambers under lower pressures, the expansion of the fluid from C to D drives the current z through the connecting ducts 8 and into the chambers comprised within the zone A—B. Fig. 9 shows a chamber of zone A—B wherein the ports 11 deliver current z, said current gradually compressing the gas until it escapes (Fig. 7) and the displacement phase starts afresh.

In Figs. 7 to 9 the two ends of ducts 8 are arranged, at opposite sides of the rotor, so as to restore the pressure-exchange current to its initial position. In practice it is possible to arrange the ducts at one side of the rotor by using one set of ports 11 (Fig. 6) only, the stream z then flowing in the chamber first in one, and then in the reverse direction.

In some cases I may advantageously determine the location of the pressure-exchange current and guide positively such current within the chambers 2 by means of partitions extending into said chambers in order more effectively to prevent intermixing of the gas currents at different temperatures, such partitions bounding in each chamber a zone more particularly adapted to receive and localize the pressure-exchange current. In a convenient embodiment of my invention, illustrated in Figs. 15 to 18 of the accompanying drawings, the said partitions, shown at 21, extend co-axially to the rotor 1, in a direction parallel to the flow of gases during the phase of displacement or scavenging.

In this form of apparatus, the general arrangement of the inlet and outlet pipes 5, 6 and 5′, 6′ is similar to that shown in Fig. 5. In each chamber is a partition 21 which divides it into a main chamber $2a$ adapted to be periodically connected with the inlet and outlet pipes 5, 6 and 5′, 6′, and an antechamber $2b$ more particularly intended for accommodating the pressure-exchange current. To this end each antechamber $2b$ is provided at one end with a port 11 for letting in or out the pressure-exchange current and at the other end with an opening 22 communicating with the adjacent chamber $2a$.

It is desirable that during the phase B—C of scavenging under high pressure the portion of the pressure-exchange current z located in said antechamber be excluded from the scavenging in order that only the warm compressed gas x be admitted to the heat-exchanger 7. For this purpose the inlet pipe 5 and outlet pipe 6 which are connected with the high-pressure heat exchanger 7 in the manner shown in Fig. 1, have a reduced height corresponding to that of the chamber $2a$ (Figs. 15 and 17), so that only the chambers $2a$ are swept by the gases during the phase B—C of displacement or scavenging under high pressure.

The low-pressure pipes 5' and 6', on the contrary, may extend over the entire height of both chambers 2a and 2b, as it is of advantage that, during the phase D—A of scavenging under low-pressure, the pressure-exchange gas shall be evacuated and caused to give off outside the apparatus the cold generated by its expansion.

It will be observed that it is advisable that the direction of flow of the gases in the pipes 5', 6' be reversed with respect to the direction of flow in the pipes 5, 6, in order that only the gas z' which has been driven out of the antechamber 2b shall always form, during the scavenging, a cushion between the warm gas x and the cold gas y, the scavenging under high pressure being so regulated as to avoid as much as possible, or desirable, the escape of this cushion from the rotor.

It will be understood, with reference to Figs. 17 and 18, that the function of the pressure-exchange current remains the same as described above. During the compression phase A—B, the pressure-exchange current z delivered into an antechamber 2b through the successive orifices 81, 82, 83, 84 of the ducts 8, gradually compresses the gas in the adjacent chamber 2a (Fig. 18, top). The relative dimensions of the chambers 2a, 2b are such that at the beginning of the scavenging phase B—C, a part Z' of the gas z has reached a position opposite the inlet 5 and separates the outflowing warm gas x from the incoming cold gas y (Fig. 17, top). During the expansion (C—D) the gas z is driven back through the openings 81', 82', 83', 84' of the ducts 8 (Fig. 18, bottom) towards the chambers in the phase of compression. Then the low-pressure scavenging (phase D—A, Fig. 17, bottom) carries away the cold gas which filled the chambers 2a and 2b and replaces them by warmer gas to be compressed.

According to a further feature of my invention, the gaseous partition constituted by the pressure-exchange current interposed between the expanding gas and the gas undergoing compression, is so extended as to complete the separation between adjacent chambers, and to avert leakages from one chamber to another along the internal periphery of the casing. I obtain this by by-passing a portion of the pressure-exchange current from the ducts 8 through the vanes or partitions 3 which are then made hollow. To this end, as illustrated by way of example in Figs. 10 to 12, the hollow partitions 3 are extended outwardly at 30, 30' respectively, between the ports 11 on one hand, and 11' on the other hand, and they are connected to forwardly offset distributing members or ports 12, 12', while the ducts 8 are provided with branches whose orifices 85, 86, 87, 88 and 85', 86', 87', 88' are offset rearwardly with respect to the normal orifices 81, 82, 83, 84 and 81', 82', 83', 84' by an angle corresponding to one chamber.

In this manner, since the ports 12' are placed into communication with the successive branches 85', 86', 87', 88' during the expansion phase, the pressure inside a vane 3 between C and D will be slightly lower than that simultaneously obtaining in the two chambers separated by said vane. A portion of the current z therefore will be by-passed, from each chamber under expansion, towards and through the vane and into the duct 8 in front of which said vane passes, said portion then joining the main body of current z in said duct.

During the compression phase, the ports 12 are connected in a like manner with the branches 85, 86, 87, 88 of ducts 8, situated rearwardly of the respective orifices 81, 82, 83, 84 of said ducts by an angle corresponding to one chamber, so that the pressure inside a vane is slightly higher than that obtaining in the two chambers adjacent the same. A portion of the current z will therefore be by-passed from each duct 8, be collected by the ports 12 of successive vanes, and flow through the latter and follow the internal periphery of the casing to reach the chambers in the compression stage.

The quantities of fluid thus by-passed and delivered from the expansion phase on one hand, and into the compression phase on the other hand, may be variable but they must be sufficient to prevent leakages between adjacent chambers.

It will be seen that in the position of the rotor illustrated in Figs. 10, 11, 12, the orifices of ducts 8 on the "compression" side are in the radial axes of the chambers fed by said ducts, whereas on the "expansion" side the orifices face partitions 3, according to a characteristic hereinbefore set forth. In meeting this requirement it is important that, at the moment under consideration, the orifices facing the partitions will be closed thereby. To this end, over a portion corresponding to the width of the ports 11 and 11' each hollow partition or vane 3 is closed outwardly by peripheral walls 31, 31' which separate successive ports and thus prevent direct communication between the orifices 81—84, 81'—84', and the interior of partitions 3. The successive ports 12, 12' are likewise separated from each other by walls 32, 32'. For the sake of clearness, the walls 31, 31', 32, 32' in Fig. 12 have only been shown in the expansion zone C—D.

In the apparatus described so far the chambers 2 have a constant volume. However during its passage into the heat exchanger 7 or 7' the gas undergoes a change of volume due to variation of temperature, and this change of volume must be compensated by an equivalent addition or withdrawal of gas. Possible leakage losses must likewise be compensated. This work of compensation involves the most important expenditure of energy in the cycle and it is effected in the case of the example under consideration, by combining a compressor rotor with the ring of constant capacity chambers under such conditions that the continuity of the pressure-exchange current is not affected.

As is shown in Fig. 13, the rotor 13 of this compressor is mounted eccentrically within the rotor 1 and comprises a plurality of radially sliding blades 14, the number of which is slightly larger than that of the vanes 3 of the rotor 1. These blades confine variable capacity chambers 15, each connected with a chamber 2 by an opening 16.

The relative displacement of blades 14 along the inner wall of rotor 1 is limited to a small alternating motion if rotors 1 and 13 rotate at the same speed. Since each chamber 2 is in constant communication with a chamber 15 it constitutes, with the latter, a chamber whereof the capacity varies during the course of a revolution between a maximum volume corresponding to that occupied by the gas at its temperature before admission into exchanger 7 and its minimum volume corresponding to that of said gas after passing through the exchanger.

In practice however, such an arrangement would suffer from the disadvantage of periodically reversing the direction of motion of each blade along the wall 1. This disadvantage may be overcome by causing the rotor 13 to revolve slightly faster than rotor 1, so that the relative velocity of the blades along wall 1, which is a sinoidal function, shall always be positive. As is shown in Fig. 14, the openings 16 formed in the inner wall of the rotor 1 are so arranged that the compensating air current is sent into that portion of each capacity which contains gas at a like temperature. In the case of Fig. 14, the opening 16 faces the end of the capacity 2 which is opposite port 11 so that the gas entering by said opening is in contact with the compressed and heated gas, and not with the gas issuing from the connecting ducts. Thus, the opening also faces the outlet pipe 6 and the gas flowing through said opening is enabled to escape without impeding the main currents in the zone B—C.

The result of this arrangement is that the current generated by the bladed rotor is added to or subtracted from the current of air displaced at constant pressure. In point of fact, it is during the stages B—C and D—A of displacement or scavenging at constant pressure, that the bladed compressor must cause the variations of volume, and the angle of advance of said compressor must be selected with that end in view. During the stages of expansion and compression, the variation in volume is caused by the connecting ducts 8.

If desired the arrangement shown likewise allows use to be made of the compensating current, to assist the pressure-exchange current in preventing the occurrence of leakages along the lines of contact of surfaces moving relatively to each other, to this end it is only necessary to provide additional openings at points 16′ so situated in the internal wall of rotor 1 as to cause a portion of the compensating current to be by-passed through the hollow vanes or partitions 3.

Since the variation of volume produced by the compensator is practically restricted to two diametrically opposed arcs, it will be readily understood that if the angular position of said arcs be modified relatively to the arcs B—C and C—D, the variation of volume obtained during the constant pressure displacement will likewise be varied. The delivery of the compensator may be modified as desired by regulating the lead of the compressor, i. e. by turning the centre 0′ of the rotor 13 through a given angle around the centre 0 of rotor 1. Means thus are available to act upon the compensating current and, through the same, upon the other gas currents, irrespective of the manner in which the speed of rotation of the apparatus may be adjusted.

Fig. 14 shows an arrangement whereby this result may be attained in a simple and convenient manner. To the rotor 1 revolving on a stationary shaft 0 is rigidly secured an internally toothed ring 17. On shaft 0 is a crank 0′, upon which rotates the rotor 13, rigidly connected to a spur wheel 18 meshing with ring 17. Rotor 1 is driven by an appropriate motor and actuates rotor 13 at a speed determined by the ratio of the gears 17 and 18. A worm wheel 19 secured to shaft 0 meshes with a worm 20. The angular position of crank 0′, i. e. the compressor lead may thus be adjusted as desired by rotating said worm. This adjustment may be effected during operation of the apparatus, as gears 17 and 18 always remain in mesh.

The constructional embodiments above set forth may of course be varied without departure from the scope of the present invention. For example circulation of the gas currents through the chambers may be directed, as aforesaid, from the centre of the rotor towards its periphery or vice versa. In either case the circulation of the currents may be maintained by the apparatus itself, acting as a fan, the use of separate fans then being superfluous. The partitions between chambers then would be defined more exactly by the term "vane" which has been used hereabove by extension, to distinguish said partitions from the other walls. On the other hand the ring of chambers, the connecting ducts and other distributing members as well as the compensator may be shaped or arranged in any suitable manner. The term "rotary apparatus" herein is intended to include any apparatus whereof one part, whether the series of chambers or the distributor, is movable relatively to the other part, whatever may be the form or the arrangement of said parts.

I claim:

1. In a process of performing conversions of thermal and mechanical energy, in which a gas is successively subjected to a variation in pressure, a heat-exchange under constant pressure, and a reverse variation in pressure, the steps of directly transmitting pressure from the gas in one stage of pressure variation to the gas in another stage of pressure variation thus creating a pressure-exchange current, maintaining and guiding said current to cause same to form a gaseous partition between the gas in said first stage of pressure variation and the gas in said second stage of pressure variation.

2. In a process of performing conversions of thermal and mechanical energy, in which a gas is successively subjected to a compression stage, a heat-exchange under constant pressure, and an expansion stage, the steps of circulating the gas in the expansion stage and the gas in the compression stage at a certain speed and in a certain direction, directly transmitting pressure from the gas in the expansion stage to the gas in the compression stage thus creating a pressure-exchange current, causing said pressure-exchange current to circulate at the same speed and in the same direction as the gas in the expansion stage and the gas in the compression stage, and causing said pressure-exchange current to form a gaseous partition between the gas in the expansion stage and the gas in the compression stage.

3. In a process as claimed in claim 2, circulating the pressure-exchange current in a closed circuit and causing it to accompany successively the gas in the compression stage and the gas in the expansion stage without partaking in the heat-exchange.

4. In a process as claimed in claim 2, by-passing a portion of the pressure-exchange current to prevent leakage due to differences in pressure at different points of the circuit.

5. In a process as claimed in claim 2, compensating the variation in gas volume due to the heat exchange by means of a current guided in the same direction and at the same velocity as the pressure-exchange current.

6. In a process as claimed in claim 2, by-passing a portion of the pressure-exchange current to prevent leakages due to differences in pressure at different points of the circuit, compensating the variation in gas volume due to the

heat exchange by means of a current guided in the same direction and at the same velocity as the pressure-exchange current, and causing a portion of the compensating current to unite and cooperate with that portion of the pressure-exchange current utilized to impede leakage.

7. In an apparatus for performing conversions of thermal and mechanical energy, the combination of a stationary part and a movable part, one of said parts comprising a plurality of chambers, the other of said parts comprising a distributor, said distributor comprising a casing surrounding said chambers, inlet and outlet pipes for said chambers and ducts for periodically connecting said chambers with each other, a heat-exchanger connected to said inlet and outlet pipes, and means for setting up and maintaining a current of gas within said chambers and through said ducts.

8. In an apparatus for performing conversions of thermal and mechanical energy, the combination of a stationary part and a rotary part, a heat-exchanger, one of said parts comprising a ring of chambers and radial partitions between said chambers, the other of said parts comprising a casing surrounding said chambers, inlet and outlet pipes adapted periodically to connect said chambers with said heat exchanger, and ducts adapted to connect chambers of a region beyond said pipes with chambers of a region in front of said pipes, the chambers in one of said regions being adapted to contain gas in the course of expansion and the chambers in the other of said regions being adapted to contain gas in the course of compression, and means for setting up and maintaining a so called "pressure-exchange" current of gas within said chambers and through said ducts from the chambers in the region corresponding to expansion to the chambers in the region corresponding to compression.

9. In an apparatus as claimed in claim 8, said ducts having end orifices of a width equal to that of the said radial partitions.

10. In an apparatus as claimed in claim 8, said ducts having end orifices so arranged and so proportioned that one extremity of a duct is occluded by one of said partitions while the opposite end of said duct is in the radial axis of one of said chambers.

11. In an apparatus as claimed in claim 8, ports for each of said chambers, said ports being adapted to cooperate with the end orifices of said ducts.

12. In an apparatus as claimed in claim 8, said inlet and outlet pipes having orifices equal to the cross-section of the chambers at right angles to the direction of the gas currents, an orientation which is tangential to the flow of the currents at the outset of admission and the end of exhaust and a cross sectional area varying according to the generally parabolic law governing delivery during the displacement stages.

13. In an apparatus according to claim 8, said inlet and outlet pipes having closure edges so shaped and so placed relativey to each other as to influence the velocity of the gas current so as to cause same to become tangential to the ring of chambers.

14. In an apparatus according to claim 8, said outlet pipes having a closure edge shaped as a nose having its point slightly spaced away from the ring of chambers and connected by a curve with a tangent to said ring.

15. In an apparatus according to claim 8, means for guiding the pressure-exchange current in said chambers in a direction parallel to the axis of said rotary part.

16. In an apparatus according to claim 8, partitions dividing each chamber of the row of chambers into a main chamber and an antechamber, each said antechamber having at one end a port adapted to co-operate with said ducts and at its other end an aperture opening into the adjacent main chamber.

17. In an apparatus as claimed in claim 8, said radial partitions being hollow, means being provided for by-passing a portion of the pressure-exchange current through said hollow partitions to prevent leakage between the chambers separated by said partitions.

18. In an apparatus as claimed in claim 8, said radial partitions being hollow, means being provided for inducing in said hollow partitions in the region corresponding to compression a pressure slightly above that obtaining in the adjacent chambers, and for inducing in said hollow partitions in the region corresponding to expansion a pressure slightly below that obtaining in the adjacent chambers.

19. In an apparatus according to claim 8, said radial partitions being hollow, distributing ports for said hollow partitions, branches on said ducts, the orifices of said branches being offset relatively to the normal orifices of said ducts, said ports being adapted to co-operate with the orifices of said branches.

20. In an apparatus according to claim 8, means for compensating the variation in volume undergone by the gas flowing through said heat exchanger.

21. In an apparatus according to claim 8, additional chambers of variable volume associated with the first-mentioned chambers, said variable-volume chambers being so connected with said first mentioned chambers as to direct thereinto a compensating current in a direction parallel to the pressure-exchange current.

22. In an apparatus according to claim 8, an eccentric rotor mounted within said ring of chambers, and radial blades slidably mounted in said rotor and forming therewith a compressor, the variable-volume chambers comprised between said blades communicating through suitable openings with the chambers of said ring of chambers.

23. In an apparatus according to claim 8, a shaft eccentric to said ring of chambers, a rotor mounted on said shaft inside said ring of chambers and forming therewith a plurality of variable-volume chambers, said variable-volume chambers communicating with the chambers of said ring of chambers, an internally toothed ring rigidly connected with said ring of chambers, a pinion rigidly connected with said eccentric shaft and meshing with said internally toothed ring.

24. In an apparatus according to claim 8, a rotor eccentrically mounted inside said ring of chambers and forming therewith a plurality of variable-volume chambers, said variable-volume chambers communicating with the chambers of said ring of chambers, and means comprising a worm wheel and worm for varying the angular position of said rotor with respect to the axis of said ring of chambers.

25. In an apparatus according to claim 8, said radial partitions being hollow, a rotor eccentrically mounted inside said ring of chambers and forming therewith a plurality of variable-volume chambers, said variable-volume chambers being connected with the chambers of said ring of

chambers and with the interior of said hollow partitions.

26. In a process of performing conversions of thermal and mechanical energy, in which a gas is successively subjected to a compression stage, a heat-exchange stage under constant pressure, and an expansion stage, the steps of circulating the gas in the expansion stage and the gas in the compression stage at a certain speed and in a certain direction, directly transmitting pressure from the gas in the expansion stage to the gas in the compression stage, thus creating a pressure-exchange current, maintaining and guiding said current to cause it to form a gaseous partition between the gas in said compression stage and the gas in said expansion stage.

27. In an apparatus for performing conversions of thermal and mechanical energy, the combination of a stationary part and a movable part, one of said parts comprising a plurality of chambers, the other of said parts comprising a distributor, said distributor comprising a casing communicating with said chambers, inlet and outlet passages for said chambers, and means for periodically establishing communication between said chambers, a heat exchanger connected to said inlet and outlet passages, and means for setting up and maintaining a current of gas within said chambers.

28. In an apparatus for performing conversion of thermal and mechanical energy, the combination of a stationary part and a movable part, a heat exchanger, one of said parts comprising a ring of chambers and radial partitions between said chambers, the other of said parts comprising a casing surrounding said chambers, inlet and outlet pasages adapted periodically to connect said chambers with said heat exchanger, and ducts adapted to connect chambers of a region beyond said passages with chambers of a region in front of said passages, and means for setting up and maintaining a current of gas within said chambers and through said ducts.

29. In an apparatus according to claim 8, an eccentric rotor mounted within said ring of chambers, and radial blades slidably mounted in said rotor and forming therewith a compressor, the variable volume chambers comprised between said blades communicating through suitable openings with the chambers of said ring of chambers, and means for varying the angular position of said rotor with respect to said ring of chambers.

30. In an apparatus according to claim 8, an eccentric rotor mounted within said ring of chambers, and radial blades slidably mounted in said rotor and forming therewith a compressor, the variable volume chambers comprised between said blades communicating through suitable openings with the chambers of said ring of chambers, and means for rotating said rotor in the same direction as and at a slightly higher speed than said ring of chambers.

31. In a process of performing conversion of thermal and mechanical energy, in which a gas is successively subjected to a compression stage, a heat exchange stage under constant pressure, and an expansion stage, the step of circulating the gas in the expansion stage and the gas in the compression stage at a certain speed and in a certain direction, directly transmitting pressure from the gas in the expansion stage to the gas in the compression stage, thus creating a pressure exchange current, excluding said pressure exchange current from the heat exchange while maintaining and guiding said current to cause it to form a gaseous partition between the gas in said compression stage and the gas in said expansion stage.

32. In a process of performing conversion of thermal and mechanical energy, in which a gas is successively subjected to a compression stage, a heat exchange stage under constant pressure, and an expansion stage, the step of circulating the gas in the expansion stage and the gas in the compression stage at a certain speed and in a certain direction in an apparatus separate from that in which the gas is subjected to the heat exchange stage, directly transmitting pressure from the gas in the expansion stage to the gas in the compression stage, thus creating a pressure exchange current maintaining and guiding said current so that it circulates with the gas in the compression stage and the gas in the expansion stage in the first mentioned apparatus to cause it to form a gaseous partition between the gas in said compression stage and the gas in said expansion stage but which is not allowed to circulate through the heat exchange apparatus.

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April 30, 1946. C. SEIPPEL 2,399,394

PRESSURE EXCHANGER

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Fig. 1.

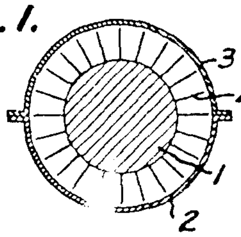


Fig. 2.

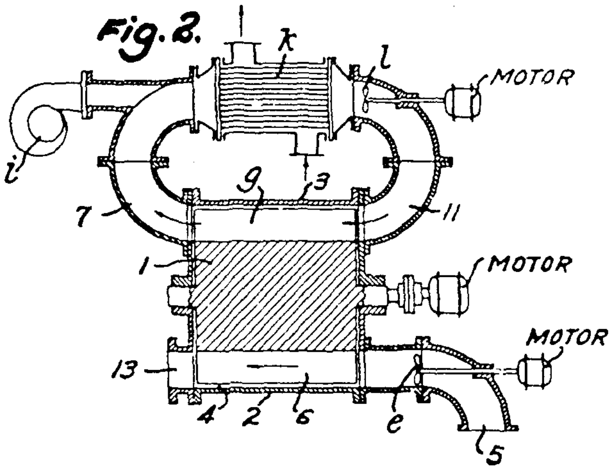


Fig. 4.

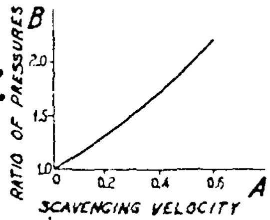


Fig. 3.

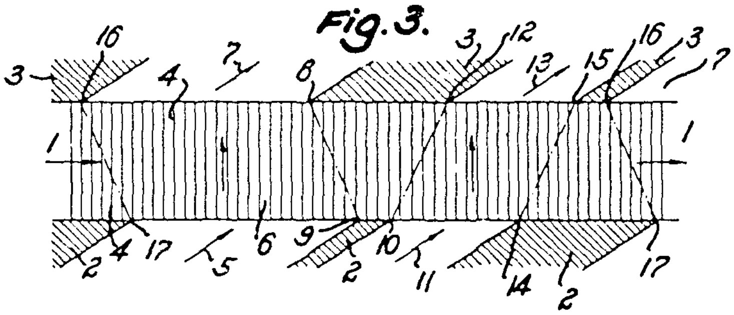


Fig. 5.

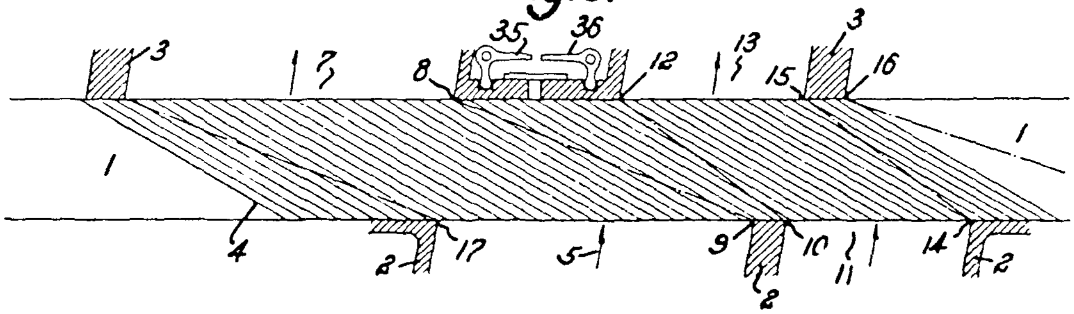


Fig. 6a.

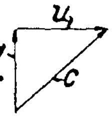
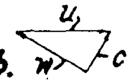


Fig. 6b.



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PRESSURE EXCHANGER

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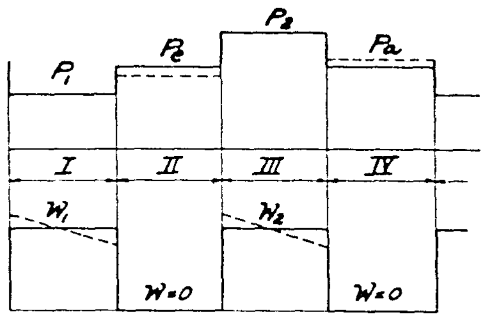

Fig. 7.

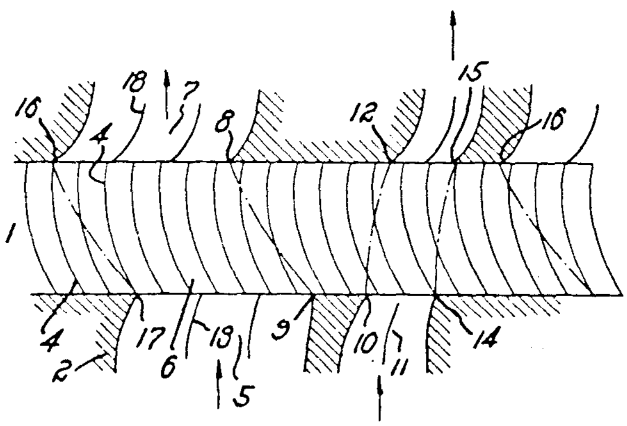

Fig. 8.

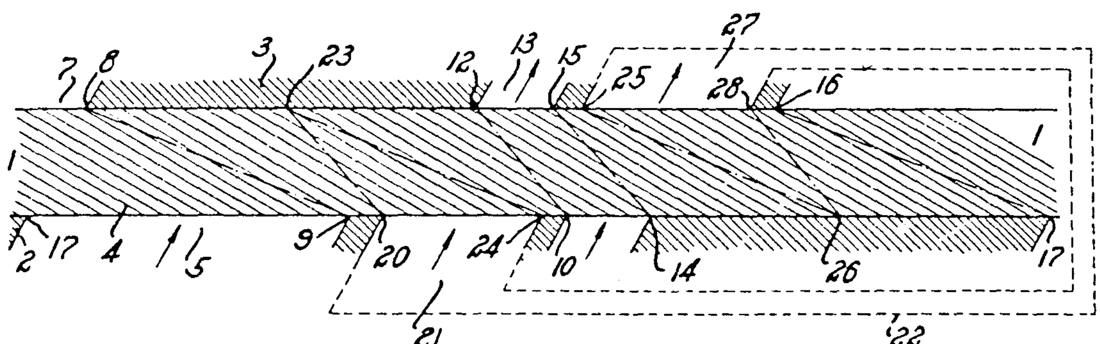

Fig. 9.

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PRESSURE EXCHANGER

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Fig. 10.

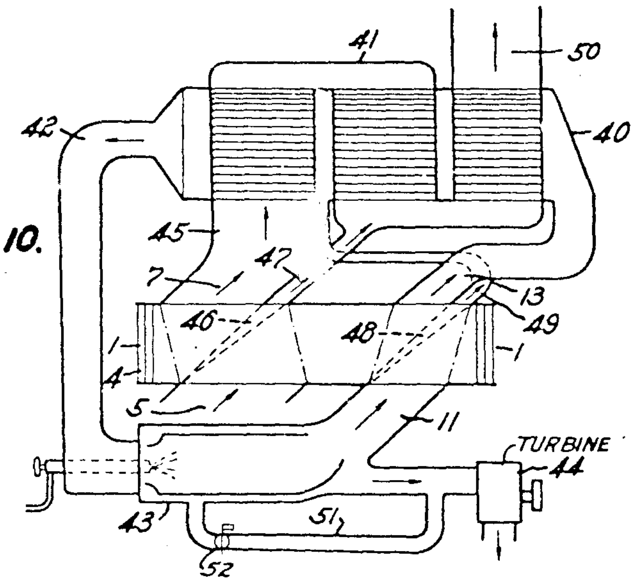


Fig. 11.

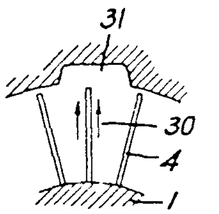


Fig. 12.

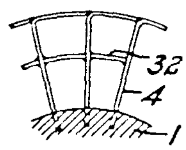
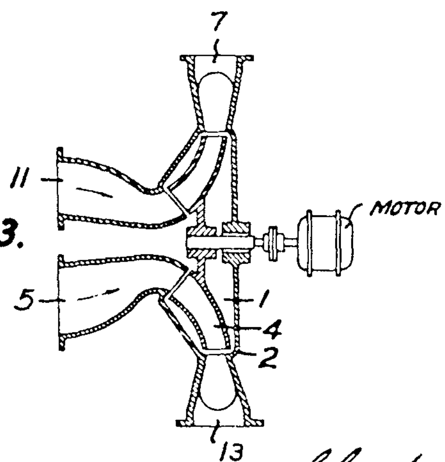


Fig. 13.



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Patented Apr. 30, 1946

2,399,394

UNITED STATES PATENT OFFICE

2,399,394

PRESSURE EXCHANGER

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Application February 2, 1942, Serial No. 429,352
In Switzerland December 7, 1940

15 Claims. (Cl. 60—41)

A pressure exchanger is a machine which takes in a gas, for instance air, at a lower pressure stage, compresses it and delivers it at a higher pressure stage, whilst at the same time the machine expands a different gas or the same gas in a different condition from the higher to the lower pressure stage. Pressure exchangers are used for refrigerators, heat pumps, gas turbines, charging sets for combustion engines, chemical processes, pressure fired steam boilers, and the like. For the two aforementioned tasks which have to be performed it is known to use cell rotors which operate in the manner illustrated in Figs. 1 and 2 of the accompanying drawings.

The present invention deals with an entirely novel construction and method of operating cell rotors and enables a machine with a good efficiency, highest capacity and compact design to be produced by effecting at least the major part of the compression by means of compression waves and at least the major part of the expansion by means of expansion waves which shoot through the cells.

The invention is explained in greater detail by means of the accompanying drawings, in which:

Fig. 1 shows a cross-section through a cell rotor and Fig. 2 one embodiment of the pressure exchanger in longitudinal section.

Fig. 3 shows the development of the periphery of a cell rotor.

Fig. 4 shows the relationship between the pressure ratio in front and behind the rotor and the scavenging velocity.

Fig. 5 shows the development of a cell rotor with helical cells.

Figs. 6*a* and 6*b* show the velocity diagrams for straight and sloping cells.

Fig. 7 is a pressure and velocity diagram for one revolution of the rotor.

Fig. 8 shows the development of a cell rotor operating as a compressor during scavenging periods.

Fig. 9 shows the development of a two-stage cell rotor.

Fig. 10 shows in diagrammatic form the application of the pressure exchanger with a combustion turbine plant.

Fig. 11 shows part of a cell rotor to an enlarged scale.

Fig. 12 shows part of a modified form of cell rotor.

Fig. 13 shows a modified form of pressure exchanger.

Figs. 1 and 2 show in diagrammatic form a cell rotor of a known type in cross-section and longitudinal section, respectively. The rotor or wheel is represented by the reference numeral 1, while 4 are the cell walls and 2, 3 the casing. Air is drawn into cell 6 through suction channel 5, for instance, by means of a fan *e*. This cell 6 after a certain rotation reaches position *g* and discharges the air into pressure chamber 7. It is assumed that the cell rotor operates as a heat pump. The compressed air is supplemented in a known manner by the air compressed in blower *i*, is cooled in a heat exchanger *k* and then by means of fan *l* is passed back to the cell rotor where it is expanded and discharged at 13.

At the instant when the compression cell comes into communication with the pressure chamber gas is impelled suddenly into the cell. When the expansion cell is opened to the lower pressure chamber gas is expelled suddenly from the former into the latter. Various means are known which serve to prevent the loss occasioned by these pulsations in the gas flow, such as eccentric location of the rotor with movable cell walls or vanes, conduits for a gradual equalization of the pressure in the compression and expansion cells, and the like. These measures certainly result in an improvement in efficiency but the capacity of machines of this kind is very limited either due to mechanical stresses or flow losses in the equalizing conduits. Furthermore, only moderate peripheral speeds and flow velocities can be obtained.

The machine consists of a simple cell rotor with fixed cell walls or vanes as shown in Figs. 1 and 2. The lower pressure stage is provided with a scavenging section in which the fresh gas to be compressed displaces the expansion gas, and in the upper pressure stage there is a scavenging section in which the gas to be expanded displaces the compressed gas. The novel method of operating the machine is achieved by the special position and shape of the fixed and movable channels in the casing, as shown in Fig. 3. This figure represents a development of the periphery of the cell rotor of one embodiment of the invention. 1—1 is the rotor in its developed form, 2—2 and 3—3 are the development of a cylindrical section through the casing on both sides of the rotor. The radial cell walls appear in the figure as straight lines 4. One rotation of the rotor corresponds to a displacement of the developed periphery from left to right. Compression gas flows from the suction space 5 into cells 6 and thus displaces the contents of the cells produced by expansion to space 7. As soon as fresh gas fills the cells the ends of the cells are closed

due to the rotation of the runner by a control edge 8 in the casing. The cell contents are still moving when the cell is closed. The sudden closing of the end of the cell produces a pressure wave the height of which depends upon the speed of the rotor and which shoots through the cell from the outlet to the inlet end. Since the cell is moving the wave front describes the dash-dot path 8—9 shown in the figure.

When the entire cell contents have reached a higher pressure level, that is at the instant when the wave front reaches the front end of the cell, it is closed by the control edge 9 so that the compressed gas is trapped and locked up to the higher pressure level.

The cell with its trapped contents moves further to the right. Its front end becomes open at 10 to space 11 in which the expansion gas is at a higher pressure than the contents of the arriving cell. This results in a fresh pressure wave which with approximately the velocity of sound shoots through the cell from the front to the rear along the path 10—12. At the instant when this pressure wave reaches the rear end of the cell it is put into communication with pressure space 13 by means of control edge 12. Behind the pressure wave the gas has begun to move with a velocity depending on the pressure jump. This flow velocity is different from the velocity of sound or the velocity of the wave front. It is generally considerably lower.

Both ends of the cell are now open and its contents are moving. The compressed gas discharges into space 13 and the gas which is to be expanded flows in from space 11, care being taken that the casing is equipped for the correct inflow and discharge conditions.

As soon as a sufficient quantity of the gas which is to be expanded has flowed in, the front end of the cell is closed by the edge 14. The supply of gas is thus interrupted suddenly and an expansion wave is generated which shoots through the cell along the path 14—15. When the expansion wave reaches the opposite end of the cell this latter is closed by edge 15. The entire contents of the cell have come to rest and the pressure is lower than that of the upper pressure stage. At the edge 16 the outlet ends of the cells open, and the contents of the cells which have come to rest commence to flow out into the space 7 until the edge 17 also uncovers the inlet ends, and this movement of the cell contents toward 7 is strengthened by the compression gas at 5 and so on. This results in a fresh expansion wave which sets the cell contents into motion again. Scavenging at the lower pressure stage is thus initiated. The cycle of operations described for the cell is completed and repeats itself. The new principle on which the machine operates is therefore useful compression by means of compression waves and useful expansion by means of expansion waves. The wandering compression and expansion waves effect a transformation between the pressure and kinetic energy of the scavenging movement.

Fig. 4 shows the relationship between the pressure ratio in front and behind the wave and the scavenging velocity. The abscissae A shows the velocity as a ratio to the velocity of sound (Mach number) whilst the ordinates B show the pressure ratio. The scavenging velocity must increase with the pressure ratio. In a rotor with axial cells the gas emerges with a velocity whose axial component is equal to the scavenging velocity and whose tangential component is equal to the peripheral speed of the rotor. At higher speeds the gases emerging from the cells possess considerable energy which can only be partially converted into useful work by means of suitable diffusors.

The outlet energy can be reduced if the axes of the cells are not arranged parallel to the axis of the rotor or in radial planes but at an angle to these planes or helically or spirally.

Fig. 5 shows the development of a pressure exchanger with a rotor having helical cells. The method of operation is fundamentally the same as that already described. Reference numerals 1 to 17 refer to the same elements as in Fig. 3.

Figs. 6*a* and 6*b* show the velocity triangles for straight and sloping cells; w is the velocity of flow relative to the cell during scavenging. This velocity determines the pressure ratio. u represents the peripheral velocity of the cell. Relative velocity and peripheral velocity give the resultant absolute velocity c. This is the velocity with which the gas emerges from the rotor into the casing. It will be noticed that in Fig. 6*a* c is considerably larger than w; in Fig. 6*b*, however, c is smaller than w.

In axial cells the pressure of the gas increases with the radius as a result of the centrifugal force. When there is a difference of density between the two gases present during scavenging, the pressure increase is greater in the gas of higher density. This upsets the equilibrium within the zone of contact of the two gases and causes the gases to mix. A strong mixture of gases is however unfavourable to the proper functioning of the heat exchange.

If however the cells are helical, the tangential component of the absolute motion is decreased and the centrifugal effect partly or totally suppressed.

Fig. 7 shows in a diagrammatic manner the course of the pressure and the velocity of flow in the center of a cell during one revolution. I is the scavenging section at the lower pressure stage with pressure P_1, and velocity w_1; II is the compression section with pressure P_2 and velocity $w=0$; III is the scavenging section at the upper pressure stage with pressure P_3 and velocity w_3; IV is the expansion section with pressure P_4 and velocity $w=0$.

The velocity of a scavenging stream which is set in motion by a pressure wave is maintained during the entire scavenging period if care is taken that the resistances in the scavenging circuit, both inside and outside the pressure exchanger, are overcome by a fan for instance. On the other hand the velocity can be allowed to decrease during the scavenging period. By this means energy is released which can overcome the resistances in the scavenging circuit. The scavenging fan is thus relieved of its load and depending upon the resistances in one or the other of the scavenging sections, it can be entirely dispensed with or the gas be used to do useful work. The corresponding course of the pressures and velocities are shown by the broken lines in Fig. 7.

On the other hand it is possible to allow the scavenging blower to produce a higher pressure than is necessary to overcome the resistances in the scavenging circuit. By this means the scavenging stream in a cell is accelerated between the beginning and end of the scavenging section. The compression wave at the end of the lower scavenging period will be increased and the gas to be compressed will be trapped at a higher pressure. Similarly at the end of the upper scav-

enging period the gas to be expanded will be trapped at a lower pressure. More gas is therefore compressed and less gas expanded. As a result of this, for instance the auxiliary blower used with a heat pump must transfer less gas and under certain conditions may even be omitted. The work to be done by the auxiliary blower is transferred to the scavenging blower.

The scavenging velocities at the lower and upper pressure stages do not need to be equal. Within certain limits it is quite safe when as a result of unequal velocities unequal pressure jumps occur. The scavenging velocities and thus also the displaced volumes can be regulated by altering the flow resistances in the scavenging circuits or by altering the pressures produced by the scavenging fans. Generally it is sufficient if one fan is provided in one of the scavenging circuits, for instance in the scavenging circuit having the higher resistance.

When the gas to be expanded has a considerably different density to that of the compressed gas (for instance when the same gas is expanded at a different temperature) the scavenging velocities at the beginning and end of the scavenging sections must be selected differently in accordance with the ratio of the sound velocities because the pressure jumps of the pressure waves depend on the relationship between scavenging and sound velocity (Mach number) as shown in Fig. 4; and the total pressure jumps on the compression and expansion side must be equal. If for instance air is compressed and expanded again at a considerably higher temperature, the scavenging velocities must decrease during the lower scavenging period and increase during the upper scavenging period, so that they are higher for the expansion wave than for the compression wave.

Under certain conditions it is possible that these alterations in scavenging velocities can be obtained without having to adopt special measures. This can be proved to be the case when in the velocity diagram of the type shown in Fig. 6a the velocities c and w are equal. When namely a light gas displaces a heavy gas the kinetic energy of the cell contents decreases in proportion to the masses when the velocity remains constant. The energy which is released serves to accelerate the scavenging stream. It is only necessary to construct the channels in the casing so that the transfer of gas from the cell rotor is as free from losses as possible. If c and w differ (Fig. 6b), the change in velocity must be assisted by external means, scavenging resistance and scavenging pressure. The changes in scavenging velocities which for various reasons are necessary or desirable overlap each other and can under certain conditions mutually balance each other.

The cell rotor described above with straight or helical cells requires only a small driving power to overcome friction losses if the gas enters the rotor parallel to the cell walls. When the gas is allowed to enter with a slight impact in the direction of rotation against the cell walls a special drive can be dispensed with. The scavenging stream in the cell rotor can be much more strongly diverted than is necessary merely to drive the cell rotor. When the momentum in the scavenging stream—i. e., the product of the flow velocity and the mass in flow—increases in the direction of rotation, the cell rotor operates as a turbo-compressor and must be driven from the shaft. The cell rotor assists or replaces the scavenging blower and can under certain conditions also render the supplementary blower 1 (Fig. 2) superfluous. When the momentum in the direction of rotation is decreased, the cell rotor operates as a turbine. The power delivered at the shaft is at the cost of the scavenging power. The scavenging velocity decreases rapidly from the beginning to the end of the scavenging period. The intermediate pressure Pe (Fig. 7) at which the gas is trapped on the compression side decreases and less gas is compressed. On the other hand the intermediate pressure Pa at which the gas is trapped on the expansion side increases and more gas expands.

Fig. 8 shows the development of a cell rotor which operates as a compressor during the scavenging periods. Reference numerals 1–17 indicate the same elements as in Fig. 3. It should be noticed that the channels in the casing provided with blades 18 have a variable direction in accordance with the increasing scavenging velocity. The cells shown in Fig. 8 may be varied in width by varying the angle of inclination of the cell walls 4. By means of a suitable choice of cell height it is possible to obtain a constant or only slightly variable cross-section of cell, such as is generally desired (compare Fig. 13).

The pressure exchangers described so far, which can be termed single-stage pressure exchangers, operate each with two compression and expansion waves. The pressure jump obtainable per pressure wave cannot be increased indefinitely. Fig. 9 shows the development of a two-stage cell rotor which operates with four compression and expansion waves each.

1 is the development of the cell rotor with inclined cells 4, whilst 2 and 3 are the casing. Scavenging at the lower pressure stage occurs from space 5 to space 7. The first pressure wave is produced at edge 8 which suddenly closes the cell. The second wave occurs at edge 20 where the cell comes into communication with space 21 in which a pressure prevails which is between the lower and upper pressure stage. This space 21 can be supplied with gas through a conduit 22, this gas being taken from the cells in the expansion section. This additional gas can also be taken from another source. The wave reaches the end of the cell at point 23 where it encounters a closed wall, is reflected and shoots through the cell from the rear to the front as the third compression wave. When the wave reaches the front end the cell it is closed by edge 24. During the total transition period of the wave along the path 20—23—24 gas flows from 21 into the cell. The fourth wave is initiated at edge 10 as with a single-stage rotor.

The expansion occurs in a similar sequence: first expansion wave 14—15 at the end of the scavenging, second expansion wave 25—26 when one end of the cell is opened to intermediate space 27 which can be in communication with space 21, reflection of wave at the closed end of the cell up to point 28 and return path 26—28 as the third wave. Whilst the wave follows the path 25—26—28, gas flows from the cell into space 27 and from there to space 21. The fourth expansion wave 16—17 occurs upon the initiation of the lower scavenging process.

Compression waves 20-23-24 and the corresponding expansion waves 25-26-28 can be repeated a number of times. By this means multi-stage pressure exchangers can be obtained. It is also possible to have a single-stage compression and a two-stage expansion, preferably when the expansion gas is hotter than the compression gas. Separate pressure exchangers can also be connected in series.

In order to explain the application of the pressure exchanger more clearly a combustion turbine plant is illustrated diagrammatically in Fig. 10. 1—1 is the developed periphery of the cell runner. Only a few of the cell walls 4 are shown in the drawings. Fresh air enters the rotor at 5, is compressed by two pressure waves and emerges at 13. Compressed air enters the three-stage heat exchanger 41 at 40 where it is further preheated and passes out again at 42. It is then heated further in a combustion chamber 43 where fuel is burnt. Part of the exhaust gases from the combustion chamber are extracted as useful air and can for instance be supplied to a turbine 44. The remainder of the gases from the combustion chamber return at 11 to the pressure exchanger where they are expanded and emerge at 7 in order to flow at 45 into heat exchanger 41, where part of their residual heat is transferred to the compressed air, the gases escaping through the flue 50.

The air entering at 5 displaces the exhaust gas emerging at 7. The line of contact of the two gases is not sharply defined. Due to heat conduction, turbulence, formation of boundary layers at the walls and differences in the mass forces in the gases of different density, there is generally an undesirable mutual penetration of the gases. The zone where this intermingling occurs extends over a wedge-shaped space 46 between the cold and warm streams 5 and 7. A similar zone 48 forms at the upper stage. This mutual penetration can be rendered to a great extent harmless when the mixing zones are scavenged and care is taken that practically only cold air is locked in and only hot gases are locked out.

Heat is transferred through the cell walls from the hot to the cold gas. The heating up of the cold gas and the cooling down of the hot gas during the short period which elapses from the time the gas flows into the cell until it is closed, is detrimental. The contents of the cell are not uniformly heated during this period but only a layer in the vicinity of the cell wall. In many cases, especially when the cells are inclined, the centrifugal force on the boundary layer is greater than on the gas core. The boundary layer flows outwards along the walls. In the scavenging section where there is no difference in pressure between adjoining cells, recesses and channels can be provided in the casing which trap the outflowing boundary layer and conduct it away with mixed gas.

Fig. 11 represents a section of a pressure exchanger. The arrows 30 indicate the boundary layer flow; 31 is the channel in the casing which serves to trap the boundary layer which is whirled out.

When, as in Fig. 10, there is a heat exchanger, it is an advantage if the scavenged mixed gases are separately trapped by channels 47 and 49 and either conducted to an intermediate stage or not at all to the heat exchanger.

In the arrangement shown in Fig. 10 an air pipe 51 with a control element 52 leads to turbine 44. Since the walls of the cell rotor are alternately in contact with cold and hot gases it can often stand a higher temperature than the turbine 44 which is operated only by hot gas. As a result of this it is necessary to be able to regulate both temperatures independently, this being possible by means of air pipe 51. Turbine 44 can also be exclusively supplied with preheated air through pipe 51 if it is for instance desired to keep it free from ashes. The turbine generally operates with higher flow velocities than the cell rotor and is more liable to erosion.

The rotor of the pressure exchanger can be built with cells which are open or closed at the periphery. Fig. 1 shows an embodiment with open cells and Fig. 12 one with closed cells. The cell walls are bent over at the ends and welded together. High cells can be subdivided by an intermediate wall 32 which takes part of the centrifugal forces acting on the cell contents. The intermediate wall can also have a continuation in the casing whereby the flow can be adjusted to the various peripheral velocities of the inner and outer cell parts. The generatrix of the body of revolution formed by the cells can be axial, diagonal or radial. It can be straight or curved. Fig. 13 shows a pressure exchanger whose cross-section is similar to that of a centrifugal blower. This shape can be used to advantage when the cell rotor operates as a turbo-compressor during the scavenging period.

From the description of the operation of the pressure exchanger it will be noted that it is important that the ends of the cells should be accurately opened at the right moment by the control edges. It is therefore an advantage to construct at least a number of the control edges so as to be adjustable. In Fig. 5 for instance, edges 8 and 12 can be displaced by means of levers 35 and 36. Their position can therefore be adjusted to suit any alterations in the velocity of sound due to a change in temperature.

Although the time required by the control edges to open the ends of the cells is small, pressure waves with a flattened front will result. The first pressure impulse which occurs when the cell is opened wanders a certain distance along the cell until its entrance is entirely free and the gas can enter without hindrance. Care must be taken that this path relative to the length of the cell is not too long; this can be achieved by a suitable choice of cell division, peripheral velocity, and angle of inclination of the cells. On the other hand when fixing these values the flow losses and the heat transfer have to be taken into account. Finally there are cases where in order to suit various operating conditions variable angles of flow are necessary, these being obtained by means of rotatable running or guide blades.

I claim:

1. A pressure exchanger comprising a rotor carrying a plurality of cells extending therethrough, a stator casing including inlet means adjacent one end of said cells for supplying a gas at one pressure stage to said cells and for supplying a gas at a high pressure to said cells at a further point along the circumference of the rotor, outlet means adjacent the other end of said cells whereby said first gas is delivered from said cells at an increased pressure and said second gas is delivered from said cells at a decreased pressure, and control surfaces in operable relation to the ends of said rotor cells, said inlet and outlet means being spaced in said control surfaces to open the inlet ends of the rotor cells to inflow of the first gas, while opening the outlet ends of the cells at a point preceding the first opening of the inlet ends of the cells by approximately the time for a pressure wave to traverse the cells, thereafter to close the outlet ends of said cells at the point where the first gas has traversed the cells for a period of approximately an even multiple of the time for a pressure wave to traverse the cells while closing the inlet

ends of the cells at a point trailing the first closure of the outlet ends of the cells by approximately the time for a pressure wave to traverse the cells, thereafter to open the inlet ends of the cells to inflow of the second gas while opening the outlet ends of the cells at a point trailing the opening of the inlet ends of the cells by approximately the time for a pressure wave to traverse the cells, and to close the outlet ends of the cells at the point where the second gas has traversed the cells while closing the inlet ends of the cells at a point preceding the second closure of the outlet ends of the cells by approximately the time for a pressure wave to traverse the cells for a period approximately an even multiple of the time for a pressure wave to traverse the cells, and thereafter to again open the inlet ends of the cells to inflow of the first gas until the second gas is scavenged by the first gas, whereby at least a substantial part of the compression of the first gas and the expansion of the second gas are effected by the generation of compression and expansion waves which traverse the cells through said rotor.

2. A pressure exchanger comprising a rotor carrying a plurality of cells extending therethrough, a stator casing including inlet means adjacent one end of said cells for supplying a gas at one pressure stage to said cells and for supplying a gas at a higher pressure to said cells at a further point along the circumference of the rotor, outlet means adjacent the other end of said cells whereby said first gas is delivered from said cells at an increased pressure and said second gas is delivered from said cells at a decreased pressure, and control surfaces in operable relation to the ends of said rotor cells, said inlet and outlet means being spaced in said control surfaces to open the inlet ends of the rotor cells to inflow of the first gas, while opening the outlet ends of the cells at a point preceding the first opening of the inlet ends of the cells by approximately the time for a pressure wave to traverse the cells, thereafter to close the outlet ends of said cells at the point where the first gas has traversed the cells for a period at least twice the time for a pressure wave to traverse the cells while closing the inlet ends of the cells at a point trailing the first closure of the outlet ends of the cells by approximately the time for a pressure wave to traverse the cells, thereafter to open the inlet ends of the cells to inflow of the second gas while opening the outlet ends of the cells at a point trailing the opening of the inlet ends of the cells by approximately the time for a pressure wave to traverse the cells, and to close the outlet ends of the cells at the point where the second gas has traversed the cells while closing the inlet ends of the cells at a point preceding the second closure of the outlet ends of the cells by approximately the time for a pressure wave to traverse the cell for a period at least twice the time for a pressure wave to traverse the cells, and thereafter to again open the inlet ends of the cells to inflow of the first gas until the second gas is scavenged by the first gas, whereby at least a substantial part of the compression of the first gas and the expansion of the second gas are effected by the generation of compression and expansion waves which traverse said cells through said rotor.

3. A pressure exchanger as defined in claim 2 including control edges in said control surfaces adjacent the inlet and outlet openings in the casing and means for adjusting the position of the control edges in a circumferential direction.

4. A pressure exchanger as defined in claim 2 wherein the cells extend axially through the rotor.

5. A pressure exchanger as defined in claim 2 wherein the cells extend helically through the rotor whereby the velocities of the gases in the inlet and outlet means in the casing are reduced.

6. A pressure exchanger as defined in claim 2 wherein the incoming gas stream enters the cells with an impact in the direction of rotation of the rotor so as to propel the rotor.

7. A pressure exchanger as defined in claim 2 wherein the cell walls are bent in a direction to cause the incoming gas stream to propel the rotor.

8. A pressure exchanger as defined in claim 2 wherein the cell walls are bent to such an extent that the momentum of the incoming gas stream at the outlet of the cell in the direction of rotation is greater than at the inlet so that the cell rotor operates as a power-absorbing turbo-blower in the scavenging section.

9. A pressure exchanger as defined in claim 2 wherein the cell walls are bent to such an extent that the momentum of the incoming gas stream at the outlet of the cell in the direction of rotation is greater than at the inlet so that the cell rotor operates as a power-absorbing turbo-blower in the scavenging section generating more pressure than is necessary to overcome the resistances at constant speed.

10. A pressure exchanger as defined in claim 2 wherein the cell walls are bent to such an extent that the momentum of the incoming gas stream at the outlet of the cell in the direction of rotation is smaller than at the inlet so that the rotor operates as a power-supplying turbine in the scavenging section.

11. A pressure exchanger comprising a rotor carrying a plurality of cells extending therethrough, a stator casing including inlet means adjacent one end of said cells for supplying a gas at one pressure stage to said cells and for supplying a gas at a high pressure to said cells at a further point along the circumference of the rotor, outlet means adjacent the other end of said cells whereby said first gas is delivered from said cells at an increased pressure and said second gas is delivered from said cells at a decreased pressure, and control surfaces in operable relation to the ends of said rotor cells, said inlet and outlet means being spaced in said control surfaces to open the inlet ends of the rotor cells to inflow of the first gas, while opening the outlet ends of the cells at a point preceding the first opening of the inlet ends of the cells by approximately the time for a pressure wave to traverse the cells, thereafter to close the outlet ends of said cells at the point where the first gas has traversed the cells for a period of approximately an even multiple of the time for a pressure wave to traverse the cells while closing the inlet ends of the cells at a point trailing the first closure of the outlet ends of the cells by approximately the time for a pressure wave to traverse the cells, thereafter to open the inlet ends of the cells to inflow of the second gas while opening the outlet ends of the cells at a point trailing the opening of the inlet ends of the cells by approximately the time for a pressure wave to traverse the cells, and to close the outlet ends of the cells at the point where the second gas has traversed the cells while closing the inlet ends of the cells at a point preceding the second closure of the

outlet ends of the cells by approximately the time for a pressure wave to traverse the cells for a period approximately an even multiple of the time for a pressure wave to traverse the cells, and thereafter to again open the inlet ends of the cells to inflow of the first gas until the second gas is scavenged by the first gas, whereby at least a substantial part of the compression of the first gas and the expansion of the second gas are effected by the generation of compression and expansion waves which traverse the cells through said rotor, said stator casing including a passage having an inlet opening to the cells beginning at a point trailing the first closure of the outlet ends of the cells by approximately twice the time for a pressure wave to traverse the cells and extending for a period of approximately twice the time for a pressure wave to traverse the cells and having an outlet opening from the cells beginning at a point preceding the second closure of the outlet ends of the cells by approximately twice the time for a pressure wave to traverse the cells and extending for a period of approximately twice the time for a pressure wave to traverse the cells whereby to bring about a second pair of pressure waves in the compression and expansion zones.

12. A pressure exchanger as defined in claim 2 wherein the scavenging section provided by the opening of the inlet ends of the cells to inflow of the first gas is of such length that at least a portion of the mixed incoming and displaced gases is scavenged.

13. A pressure exchanger as defined in claim 2 wherein channels are provided in the casing to trap the boundary layers of gas centrifugally projected by the cell walls.

14. A pressure exchanger as defined in claim 2 wherein the casing is provided with separate channels for scavenging at least a portion of the mixed incoming and displaced gases.

15. A pressure exchanger as defined in claim 2 including means for heating the higher pressure gas, a turbine for expanding a portion of the compressed gas, and conduit means for separately passing portions of the heated, compressed gas to the turbine and to the pressure exchanger.

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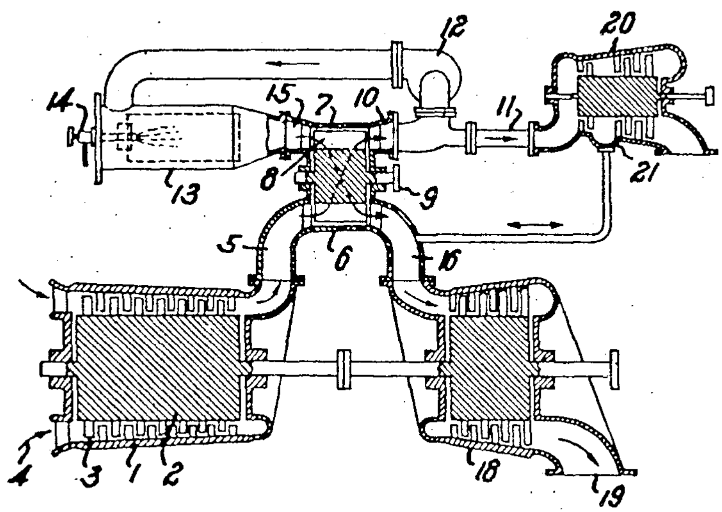
GAS TURBINE INSTALLATION

Filed March 16, 1943 2 Sheets–Sheet 1

Fig. 1.



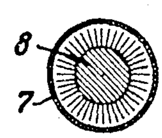


Fig. 2.

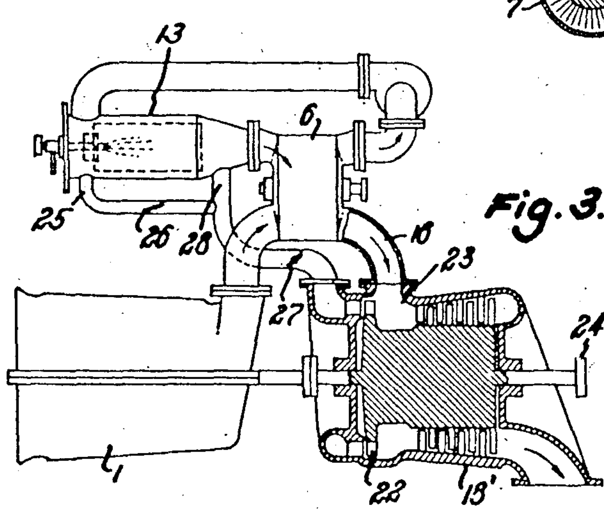


Fig. 3.

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GAS TURBINE INSTALLATION

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Fig. 4.

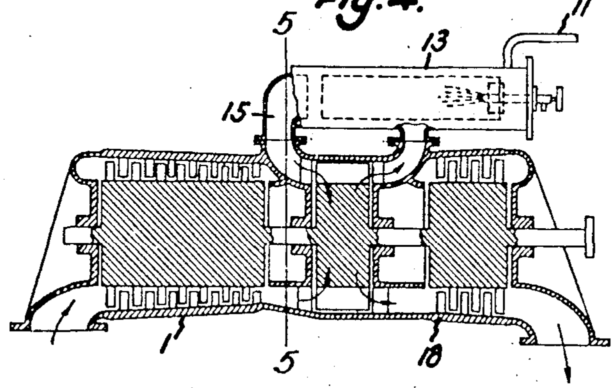
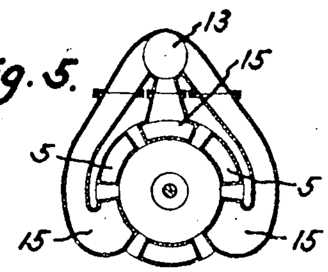


Fig. 5.



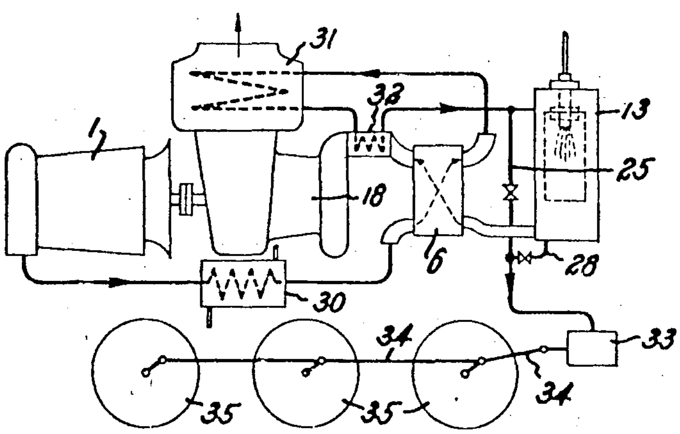


Fig. 6.

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2,461,186

GAS TURBINE INSTALLATION

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Application March 16, 1943, Serial No. 479,369
In Switzerland February 20, 1942

Section 1, Public Law 690, August 8, 1946
Patent expires February 20, 1962

13 Claims. (Cl. 60—41)

1

This invention relates to combustion gas turbine plants of the type in which the gas turbine drives a compressor for force air into a combustion chamber in which fuel is burned to develop the gaseous pressure medium that drives the turbine. This type of turbine plant was first proposed many years ago but the construction of operative turbine plants was delayed until recent years as the gas turbines and air compressors previously available were of such low efficiency that the power required for driving the air compressor exceeded the power output of the turbine.

The efficiency of the air compression is of paramount importance as regards the practical results which can be obtained with a gas turbine. It is therefore an advantage to use turbo-compressors for this purpose, especially of the axial type, because these attain a very high efficiency.

It is known to compress the air in a so-called cell-runner pressure exchanger which combines compression of the air with expansion of the combustion gases. The cells of the pressure exchanger trap the air which is to be compressed from a space which is under a higher pressure and sluice out the working gas which is to be expanded into another space.

The gas turbine installation according to the present invention consists of the combination of a turbo-machine with at least one cell-runner pressure exchanger for the upper stage. The advantage of this combination is that with the cell-runner pressure exchanger higher compression pressures can be obtained in a more economical manner than if the entire compression is accomplished solely by means of turbo-compressors. Cell-runner pressure exchangers have namely the advantage that compression and expansion are effected in the same machine. Furthermore they are mechanically simpler than turbo-compressors, can be used for much higher temperatures than ordinary gas turbines and also enable relatively small volumes to be dealt with at a high efficiency thus making them very suitable for a high-pressure stage.

Constructional examples of the invention are illustrated diagrammatically in the accompanying drawing where:

Fig. 1 shows a gas turbine installation in longitudinal section.

Fig. 2 shows a cross-section of the pressure exchanger of Fig. 1,

Fig. 3 shows a modified form of the invention,

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Fig. 4 shows a further modified form of the installation shown in Fig. 3,

Fig. 5 shows a cross-section on the line 5—5 of Fig. 4 and

Fig. 6 shows a gas turbine installation for use in a locomotive.

In Fig. 1 which shows in diagrammatic form a gas turbine installation mostly in sectional view, 1 represents a pre-compressor constructed as a multi-stage axial blower with the rotor 2; 3 are the first blade rows. Air enters at 4 and passes at the point 5 to the cell-runner pressure exchanger 6, subsequently briefly referred to as the pressure exchanger. Fig. 2 shows a cross-sectional view of the pressure exchanger, which consists of the housing 7 and the cell-runner 8 of known type with a plurality of helically arranged blades. The runner can be rotated by the shaft 9 or kept in rotation by the air and gas flow. The air entering at 5 is trapped in the cells and conveyed to the opposite side of the housing, where it emerges at 10.

Part of the compressed air flows through conduit 11 to the turbine 20 where it is employed for doing useful work whilst the remaining part of the air is conveyed by the fan 12 to the combustion chamber 13 where it supplies the necessary oxygen in the form of combustion air for the fuel supplied at 14 and also an additional quantity that serves as cooling air flowing around the combustion chamber, finally mixing with the combustion gases at the end of the combustion chamber. This stream enters the pressure exchanger 6 at 15 where it is pre-expanded, and then emerges from the pressure exchanger at 16, being finally expanded in the turbine 18, and subsequently escaping to the atmosphere at 19 or being employed further in a heat exchanger.

The turbine 20 is of the multistage type and has an interstage section or bleeding point 21 which is connected with the gas discharge conduit 16 from the pressure exchanger 6. For the purpose of a correct power distribution air can either flow from the turbine 20 to the compressor set or gas can pass in the opposite direction, as indicated by the double arrow in the figure.

Fig. 3 illustrates a further application of the invention. The installation illustrated comprises again a turbo-compressor 1', a pressure exchanger 6, a combustion chamber 13 and a turbine 18'. Instead of as in Fig. 1 extracting useful air at 11, in this case combustion gas is by-passed at 28 and supplied to a preliminary stage 22 of the turbine 18'. In the space 23 the gas which is taken away at 28 mixes with the portion which flows

through the pressure exchanger and together with this latter performs work in the lower stages. The useful work done by the plant is obtained at the shaft 24.

By extracting air at 25 the temperature of the gases tapped off at 28 can be reduced to a value which is allowable for the turbine. The temperature can be regulated by means of a valve 26. The gases which flow in at 16 can be hotter than those coming from the high pressure portion 22 so that the gases of the preliminary stage 22 are subjected to an intermediate heating. The partial amount flowing through the high pressure part of the turbine can be regulated by an element 27 which can be in the form of a throttle valve, nozzle valve, rotatable guide blade and the like. The high pressure part of the turbine can consist of one or more stages. It can be located in the main casing or in a separate casing.

With the arrangement illustrated in Fig. 4 the compressor 1, pressure exchanger 6 and turbine 18 are arranged on the same shaft. Fig. 5 shows a cross-section on the line 5—5 in Fig. 4. Air passes through the three conduits 5 from the precompressor 1 into the runner of the pressure exchanger 6. In contrast to the arrangement previously described each cell of the pressure exchanger passes through the compression and expansion cycle three times during a revolution. The air enters the combustion chamber 13 and the gases or part of the gases pass through the three conduits 15 back to the pressure exchanger 6. Dividing the circumference in the pressure exchanger into several working cycles has the advantage that the radial forces acting on the runner are balanced.

Fig. 6 shows a power installation according to the invention for a locomotive. The air is precompressed in the compressor 1, cooled in an intermediate cooler 30, further compressed in the pressure exchanger 6 and heated in the exhaust gas preheater 31. Due to the high gas temperatures which the pressure exchanger can stand the air can be further heated at 32 by the gases which leave the pressure exchanger. Part of the gases or the air is extracted at 28 or 25 and passed to the working cylinder 33 of the locomotive.

The invention can of course also be realized in practice in a variety of other ways. In addition to the advantages already mentioned, the following are also obtained.

The characteristic feature of the pressure exchanger is that comparatively cold air and hot gas are respectively compressed and expanded in the same runner. The runner thus attains a temperature which is the mean of the temperatures of the two media. This enables the pressure exchanger to be capable of working with very hot gases. An intermediate heating of the gases at the entrance to the low-pressure stage can thus be avoided.

A further feature of the pressure exchanger is that a certain amount of heat is transmitted from the hot gases to the colder air, whilst this heat flow is prevented when compressing and expanding in separate machines, as is the case in the low-pressure stage.

Heating the air at the low-pressure stage is, however, much more detrimental than at the high-pressure stage, because it either increases the work involved by the subsequent compression or necessitates the removal of this heat in an intermediate cooler. The heat received at the high-pressure stage on the other hand remains in the circuit.

The heat given off by the gases in the pressure exchanger appears at first sight to be a disadvantage because it reduces the working capacity of the turbine. The special properties of the pressure exchanger enable the temperature of gases entering the pressure exchanger, however, to be increased to such an extent that despite the heat which is given to the air the gases passing to the low-pressure stage still have the temperature required for the turbine, so that there is actually no disadvantage from this arrangement.

I claim:

1. A combustion gas turbine plant of the type including a combustion chamber, a turbine operating on combustion gases developed in said chamber, means for supplying fuel to said combustion chamber, and multi-stage compressor means for supplying compressed air to said combustion chamber; characterized by the fact that a high pressure stage of said compressor means and a preliminary expansion stage for combustion gases from said combustion chamber is constituted by a single cell-runner pressure exchanger comprising a relatively rotary cell-runner and housing therefor, said housing having a set of inlet and outlet openings for the air to be compressed and another set of inlet and outlet openings for the combustion gases to be expanded, said cell-runner including a cylindrical assembly of longitudinally extending cells into which air and combustion gases are alternately admitted and discharged during relative rotation of said cell-runner and said housing, passage means extend from said air inlet and outlet openings to connect the pressure exchanger between a low pressure stage of the compressor means and the combustion chamber, passage means connects the combustion chamber to the combustion gas inlet of the pressure exchanger, and passage means connects the combustion gas outlet of the pressure exchanger to said turbine.

2. A combustion gas turbine plant as recited in claim 1, wherein a low pressure stage of said compressor means is a turbo-compressor.

3. A combustion gas turbine plant as recited in claim 1, wherein a conduit extends from the air outlet of said pressure exchanger to a load device.

4. A combustion gas turbine plant as recited in claim 1, wherein a low pressure stage of said compressor means is driven by said turbine.

5. A combustion gas turbine plant as recited in claim 1, wherein a conduit extends from said combustion chamber to by-pass pressure fluid around said pressure exchanger to a point of use.

6. In a combustion gas turbine plant, a combustion chamber, means for supplying fuel to said chamber, a multistage air compressor means for supplying compressed air to said combustion chamber, a turbine having an inlet connected to said combustion chamber, a second turbine of multistage type having an inlet connected to the outlet of said multistage air compressor means, and a bleeder connection from an intermediate stage of said second turbine to the inlet of the first turbine.

7. In a combustion gas turbine plant as recited in claim 6, wherein a high pressure stage of said air compressor means comprises a cell-runner pressure exchanger and passage means connecting the same between the combustion chamber and the first turbine.

8. In a combustion gas turbine plant, a combustion chamber, means for supplying fuel to said combustion chamber, compressor means for sup-

plying air to said combustion chamber, and a gas turbine connected to and driving at least one stage of said compressor means; a high pressure stage of said compressor means being constituted by a cell-runner pressure exchanger comprising a relatively rotary cell-runner and housing therefor, said housing having a set of inlet and outlet openings for the air to be compressed and another set of inlet and outlet openings for the combustion gases to be expanded, said cell-runner including a cylindrical assembly of longitudinally extending cells into which air and combustion gases are alternately admitted and discharged during relative rotation of said cell runner and said housing, passage means extend from said air inlet and outlet openings to connect the pressure exchanger between a low pressure stage of the compressor means and the combustion chamber, passage means connects the combustion chamber to the combustion gas inlet of the pressure exchanger, and passage means connects the combustion gas outlet of the pressure exchanger to said turbine.

9. In a combustion gas turbine plant, a combustion chamber, means for supplying fuel to said combustion chamber, compressor means including a low pressure turbo-compressor stage and a high pressure cell-runner pressure exchanger stage, multistage turbine means, conduit means for delivering combustion gases to an intermediate stage of said turbine means through said pressure exchanger, and conduit means connecting said combustion chamber to a low pressure stage of said turbine means to supply pressure gas thereto in by-pass relation to said pressure exchanger.

10. In a combustion gas turbine, the invention as recited in claim 9 wherein said second conduit means includes a valved connection to the compressed air inlet end of said combustion chamber.

11. In a combustion gas turbine, the invention as recited in claim 9 wherein said second conduit means includes a valved connection to the combustion gas outlet end of said combustion chamber.

12. In a combustion gas turbine, the invention as recited in claim 9 wherein said second conduit means includes valved connections to the compressed air inlet end and to the combustion gas outlet end of said combustion chamber.

13. A combustion gas turbine plant comprising a multistage air compressor and a turbine on a common shaft, a combustion chamber, means for supplying fuel to said combustion chamber, conduit means connecting the high pressure stage of said air compressor to said combustion chamber, and conduit means connecting said combustion chamber to the turbine to supply combustion gas thereto; the high pressure stage of said air compressor being a cell-runner pressure exchanger connected by said conduit means between said combustion chamber and respectively said turbine and a low pressure stage of said air compressor.

CLAUDE SEIPPEL.

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Oct. 24, 1950 G. DARRIEUS 2,526,618

PRESSURE EXCHANGE APPARATUS

Filed July 29, 1947

Fig. 1.

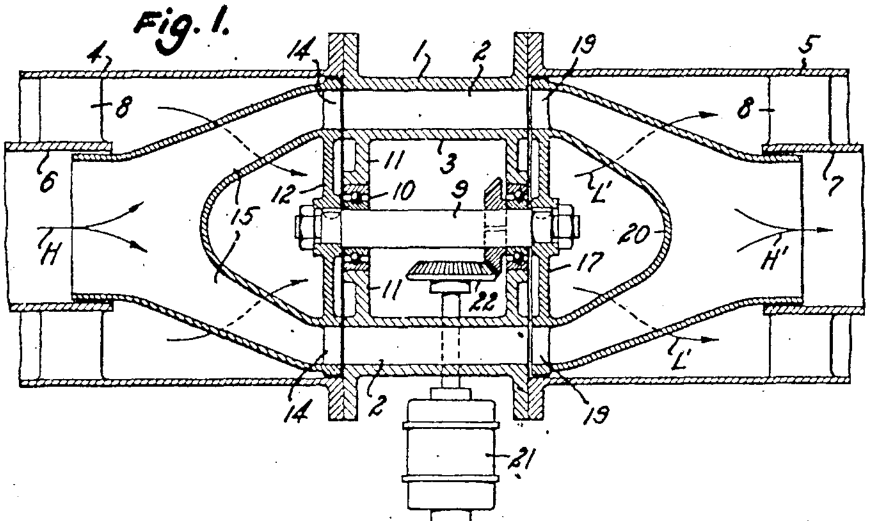


Fig. 2.

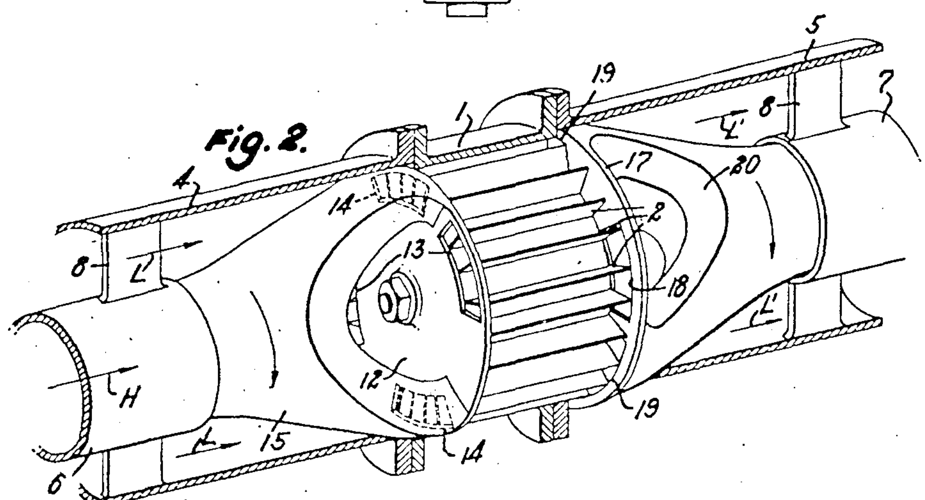
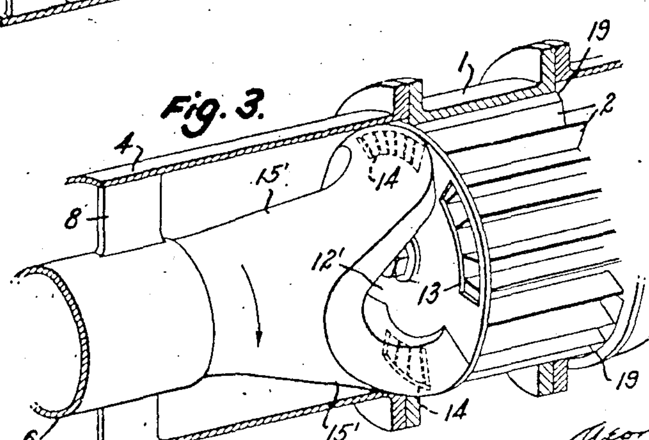


Fig. 3.



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UNITED STATES PATENT OFFICE

2,526,618

PRESSURE EXCHANGE APPARATUS

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Application July 29, 1947, Serial No. 764,499
In Switzerland July 29, 1946

8 Claims. (Cl. 230—1)

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This invention relates to pressure exchange apparatus of the type in which streams of gas under different pressures flow in alternation through the cells of a cylindrical assembly of cells or passages, the high pressure stream being brought to a lower pressure while the low pressure stream is brought to a higher pressure.

Pressure exchange apparatus of this type is well known and comprises an assembly of cells rotating between fixed end walls having inlet and outlet openings for controlling the flow of the respective gas streams through the cells in succession. The known rotary pressure exchangers have various advantages over other apparatus, for example a turbo-compressor, which could be employed to effect a pressure exchange between two gas streams but have not attained the efficiency which might be expected from an analysis of the operating conditions.

It has been proposed to take advantage of the compression and expansion waves which are set up in the several cells in succession by so locating the openings in the stationary end walls that each end of a cell is closed substantially at the instant of the arrival at that cell end of a compression or an expansion wave. The resulting increase in efficiency was substantial but was still much less than had been anticipated. I have identified one factor which lowers the predicted efficiency of a rotary pressure exchanger as the centrifugal force which disturbs the travel of pressure and expansion waves along the cells, thereby leading to losses.

Objects of the present invention are to provide pressure exchange apparatus of the cellular type which substantially eliminate losses from centrifugal force, and which thereby operate with increased efficiency. An object is to provide pressure exchange apparatus in which the cellular assembly is stationary and the flow of gases therethrough is controlled by rotating end walls or distributors. More specifically, objects are to provide pressure exchangers of the type stated which include a stationary and cylindrical array of cells, and apertured end walls or distributors which are rotated by a motor or by the flowing streams of gas.

These and other objects and the advantages of the invention will be apparent from the following specification when taken with the accompanying drawing in which:

Fig. 1 is a central longitudinal section through a pressure exchanger embodying the invention;

Fig. 2 is a perspective view of the same, with parts of the casings broken away to show the cell assembly and the distributors; and

Fig. 3 is a fragmentary perspective view of a modified form of rotary distributor.

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In the drawing, the reference numeral 1 identifies the casing or outer peripheral wall of a cylindrical array of cells formed by longitudinal partition walls 2 which extend radially from the outer wall 1 to an inner cylindrical wall 3. The casing 1 is located between and axially alined with the cylindrical inlet and discharge casings 4, 5, respectively, these casings having coaxial cylindrical conduits 6, 7, respectively secured therein by a plurality of radial webs 8. The low pressure gas stream in the inlet conduit 4 is indicated by the arrows L, and the high pressure gas stream in the coaxial inlet conduit 6 is indicated by the arrow H. The low pressure gas is compressed within the cells and discharged as a stream of annular cross-section within conduit 5, as is indicated by the arrows L', and the expanded stream of high pressure gas flows off through conduit 7, as indicated by the arrows H'.

The flow of gas through the cells is controlled by rotating distributors keyed to a shaft 9 which is located at the axis of the casing 1 and journalled in bearings 10 mounted in the end walls or flanges 11 which extend inwardly radially from the peripheral wall 3 of the cell assembly. The inlet distributor comprises a disk 12 provided with a pair of diametrically located openings 13 for admitting lower pressure gas to the cells and a second pair of diametrically located openings 14 for admitting high pressure gas to the cells, the openings 14 being the outlet ends of a bifurcated conduit 15 having a cylindrical inlet end rotatable in the outlet end of the high pressure supply conduit 6. The exhaust distributor is of similar construction and comprises a disk 17 with a pair of outlet openings 18 for the compressed gas and a pair of outlet openings 19 for the expanded gas, the openings 19 being the inlet ends of a bifurcated conduit 20 which terminates in a cylindrical section rotatable in the inlet end of the conduit 7.

The distributor disks 12, 17 are secured to the shaft 9 which is rotated by a motor 21 through gearing 22. The high pressure gas streams enter the cells with a radial component when, as shown in Figs. 1 and 2, the side walls of the bifurcated passages of the distributors are parallel to the axis of the cell assembly. This rotary component may be eliminated by imparting a helical shape to the bifurcated passages or, alternatively, the motor may be omitted when the bifurcated passages are of helical shape to develop a rotative force from the flowing stream of high pressure gas. A distributor of this type, as illustrated in Fig. 3, comprises a disk 12' and bifurcated conduit 15' of helical form.

The same cycle of operations takes place in the several cells in sequence, and there are two com-

plete cycles for each rotation of the illustrated distributors which are provided with two sets of openings for each gas stream. Two or more sets of openings are desirable for pressure and mass equalization but it is possible to operate with only one set of openings. Assuming that the cycle starts with low pressure gas entering the cell through an opening 13 of the inlet distributor 12, and the other end of the cell open to the expanded gas conduit 7 through an outlet opening 19 of distributor disk 17 and the conduit 20, the low pressure gas forces the contents of the cell (consisting of an expanded body of the high pressure gas) out at the rear of the cell. The discharge end of the cell is closed by the distributor disk 17 when the discharge of the expanded gas is completed, thereby setting up a compression wave which moves forwardly in the cell and brings the contents of the cell to rest. The rotating distributor disk 12 closes the inlet end of the cell upon the arrival of the compression wave at the inlet end, thus trapping gas within the cell under higher pressure than that of the entering stream of low pressure gas. Further rotation of the distributors brings an inlet 14 into line with the cell, and high pressure gas flows into the cell. This sets up a compression wave which travels towards the discharge end of the cell and sets the body of gas in motion. An outlet opening 18 moves into alinement with the cell as the compression wave reaches its discharge end, and the gas to be expanded pushes the compressed gas in front of it and into the compressed gas conduit 5. The inlet end of the cell is closed by distributor disk 12 when the scavenging of the compressed gas from the cell is completed, thus setting up a decompression wave which travels forwardly through the cell to bring the contents to rest. Upon the arrival of the decompression wave at the discharge end of the cell, the distributor disk 17 closes the cell which now contains an expanded body of originally high pressure gas. An outlet opening 19 then moves into alinement with the cell, and the expanded gas is discharged through conduit 20 into conduit 7, thus setting up a decompression wave which travels towards the inlet end of the cell. The cycle is completed upon the arrival of this wave at the inlet end of the cell, and an outlet opening 13 of distributor disk 12 moves into line with the cell to initiate another working cycle.

Pressure exchangers constructed in accordance with the invention operate with high efficiency as the centrifugal forces set up in the rotary distributors have little or no influence upon the travel of the compression and expansion waves along the cells.

The apparatus as illustrated and described is the presently preferred form of the invention but it is to be understood that various changes may be made in the mechanical constructions and arrangements of the parts without departure from the spirit of my invention as set forth in the following claims.

I claim:

1. Pressure exchange apparatus comprising a cylindrical assembly of cells extending longitudinally of and about an axis, a high pressure and a low pressure gas conduit at the inlet end of said cell assembly, discharge conduits for expanded high pressure gas and compressed low pressure gas at the opposite outlet end of said cell assembly, means including distributors at each end of said cell assembly for controlling the connec-

tion of the respective ends of the cells in succession to said conduits to generate compression and expansion waves in the cells when connected to the low pressure and high pressure conduits respectively, and means supporting said cell assembly and said distributors for relative rotation about said axis; characterized by the fact that said high pressure conduit is within and coaxial with said low pressure conduit, and said expanded high pressure gas conduit is within and coaxial with said compressed low pressure gas conduit.

2. Pressure exchange apparatus as recited in claim 1, wherein said cell assembly is stationary and said distributors are supported for rotation.

3. Pressure exchange apparatus as recited in claim 1, wherein said cell assembly is stationary and said distributors are supported for rotation, said distributors including helically arranged gas passages for connecting the ends of said cells to said high pressure gas conduit and said expanded high pressure gas conduit respectively.

4. Pressure exchange apparatus as recited in claim 1, wherein said cell assembly is stationary and said distributors are supported for rotation, said distributors including helically arranged gas passages for developing rotative forces from the gas stream flowing through the same.

5. Pressure exchange apparatus as recited in claim 1, wherein said cell assembly is stationary and said distributors are supported for rotation, in combination with motor means for rotating said distributors, said distributors including helically arranged gas passages for eliminating radial components from the gas streams delivered to said cells.

6. A pressure exchanger comprising a cylindrical assembly of cells extending longitudinally of and about an axis, a pair of inlet and of discharge conduits for two streams of gas under different pressures, said inlet and said discharge conduits being respectively at opposite sides of said assembly of cells and the conduits of each pair being coaxial, and means comprising rotary distributors for controlling the connection of each cell in succession to the several conduits, each distributor having a plurality of openings for connecting each cell a plurality of times to each of said conduits in alternation for each rotation of the distributors.

7. A pressure exchanger as recited in claim 6, wherein the higher pressure conduit is within the lower pressure conduit at the entrance side of said cylindrical assembly of longitudinally extending cells.

8. A pressure exchanger as recited in claim 7, wherein the discharge conduit for the expanded gas of initially higher pressure is located axially within the discharge conduit for the compressed gas of initially lower pressure.

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March 13, 1956 A. W. HUSSMANN 2,738,123

PRESSURE EXCHANGER WITH COMBINED STATIC AND DYNAMIC PRESSURE EXCHANGE

Filed Oct. 25, 1949 4 Sheets-Sheet 1

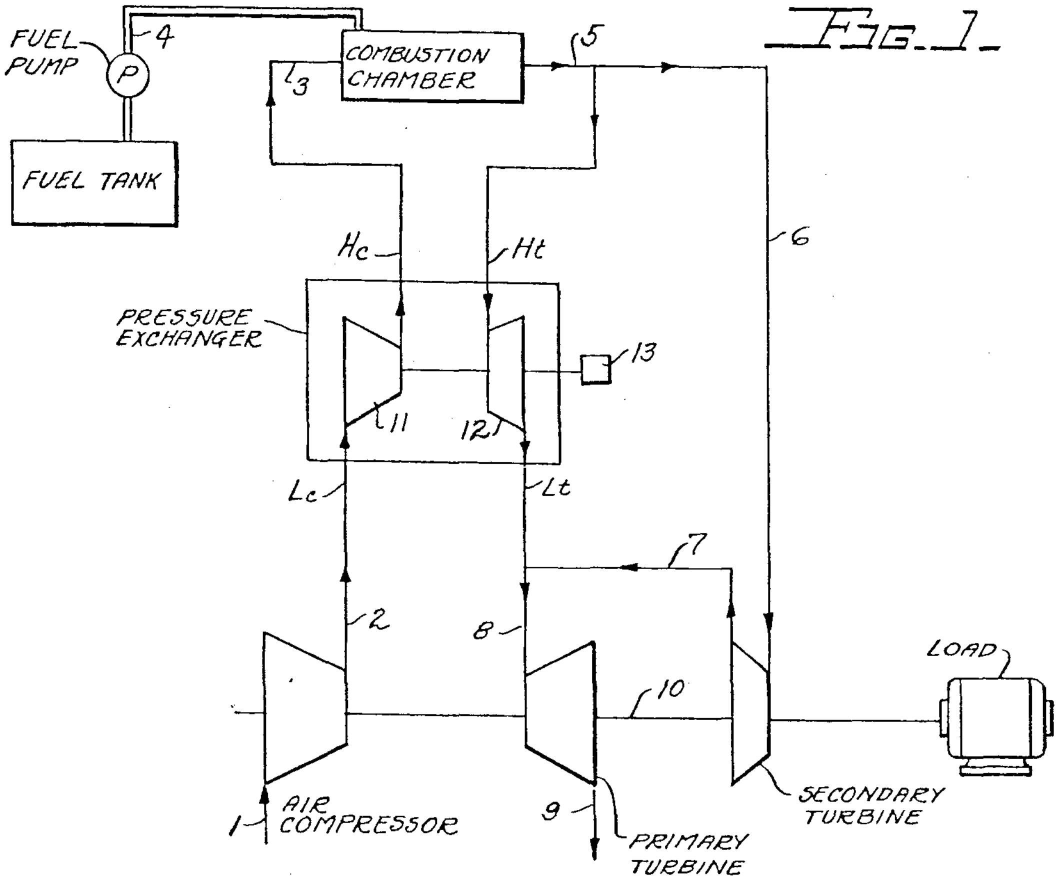

FIG. 1

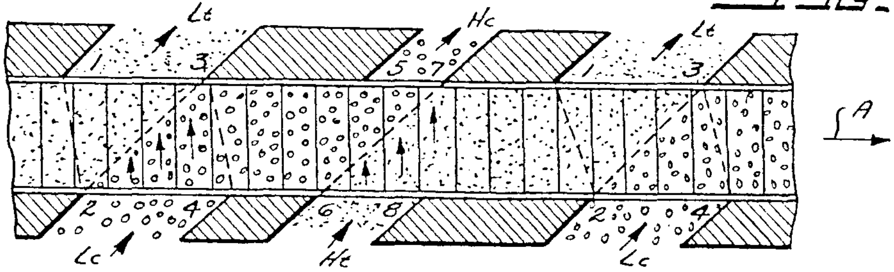

FIG. 3

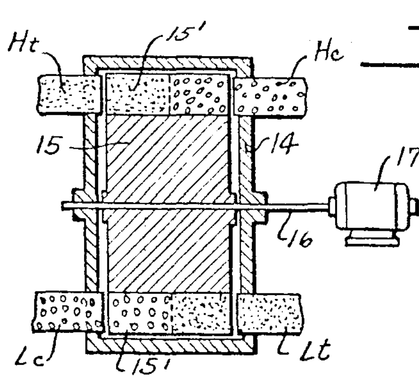

FIG. 2

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AND DYNAMIC PRESSURE EXCHANGE
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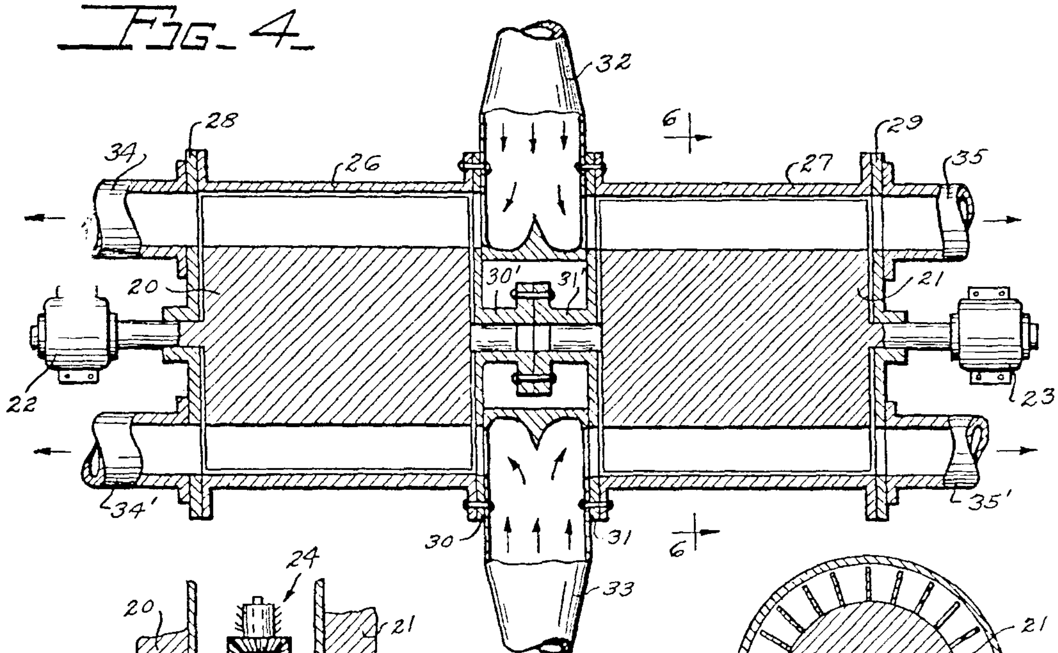
FIG. 4



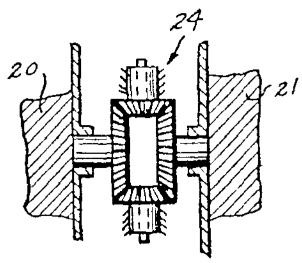


FIG. 5

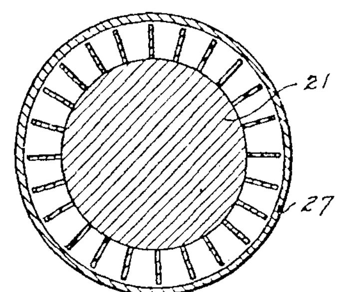


FIG. 6

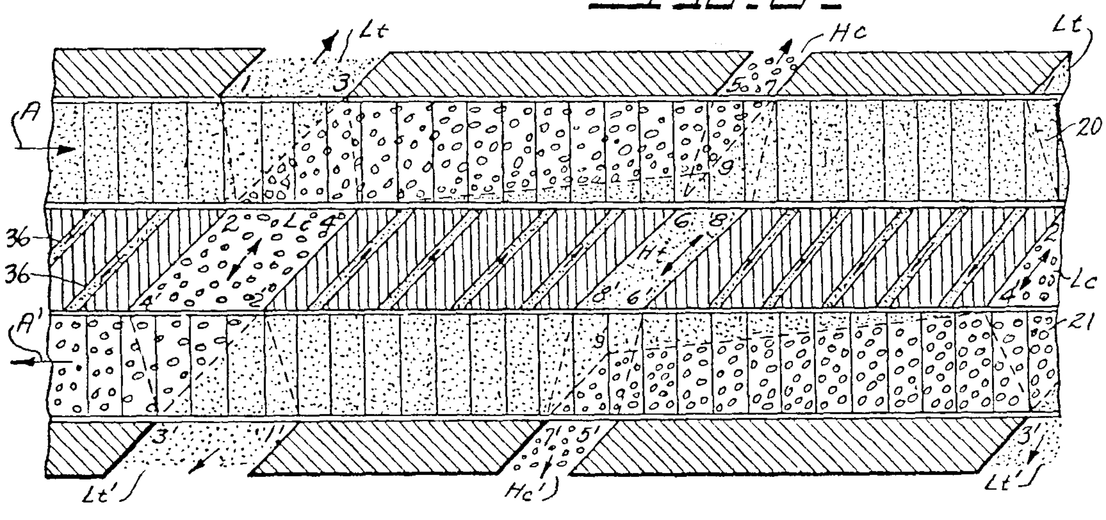


FIG. 7

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PRESSURE EXCHANGER WITH COMBINED STATIC
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Fig. 8

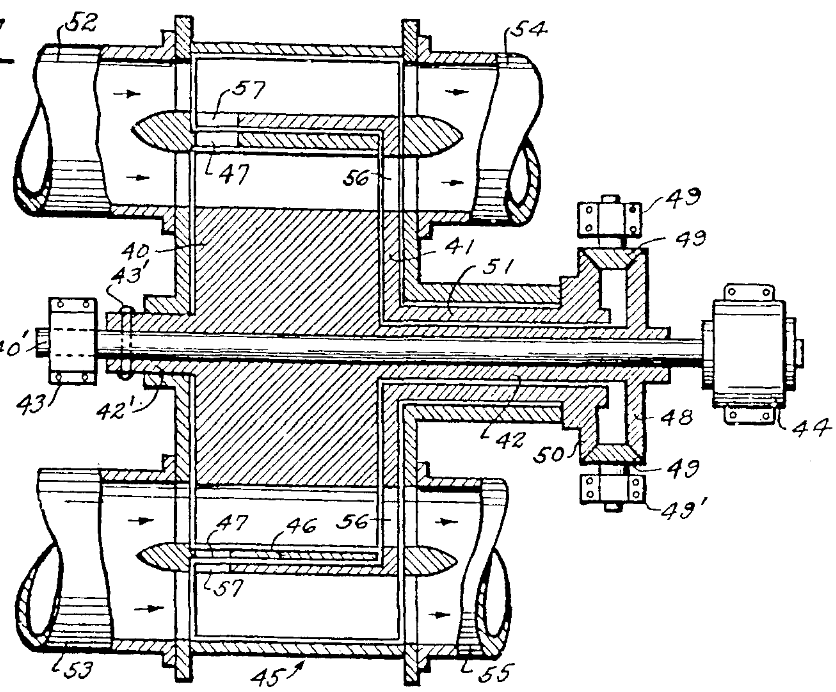


Fig. 9

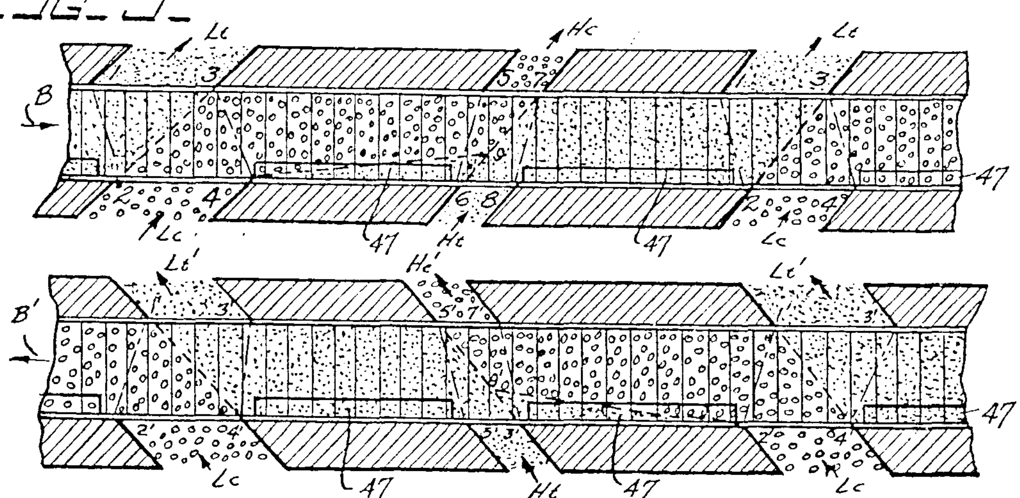
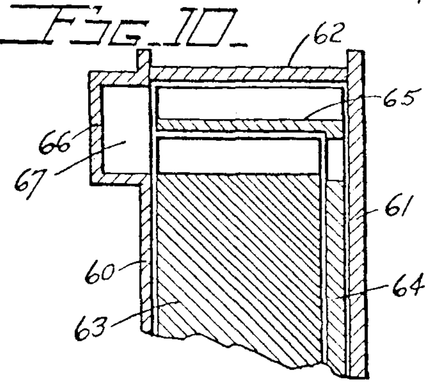


Fig. 10



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PRESSURE EXCHANGER WITH COMBINED STATIC
AND DYNAMIC PRESSURE EXCHANGE
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FIG. 11

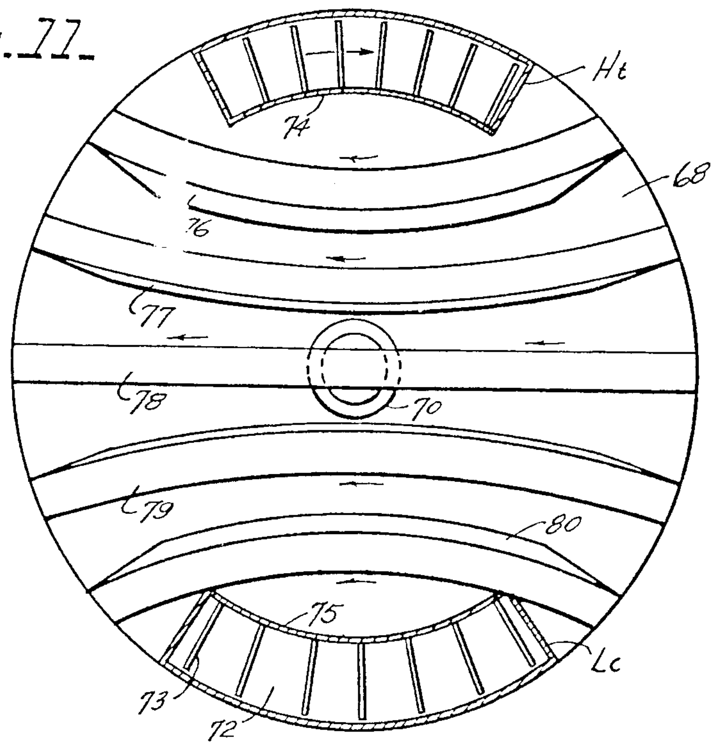
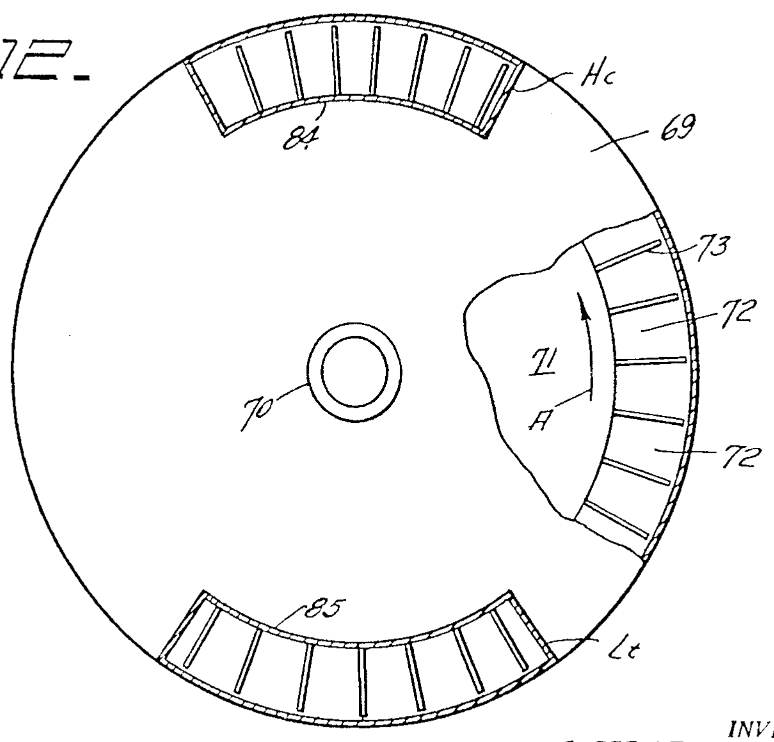


FIG. 12



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United States Patent Office

2,738,123

Patented Mar. 13, 1956

1

2,738,123

PRESSURE EXCHANGER WITH COMBINED STATIC AND DYNAMIC PRESSURE EXCHANGE

Albrecht W. Hussmann, Raleigh, N. C., assignor to the United States of America as represented by the Secretary of War

Application October 25, 1949, Serial No. 123,495

4 Claims. (Cl. 230—69)

(Granted under Title 35, U. S. Code (1952), sec. 266)

The invention described herein may be manufactured and used by or for the United States Government for governmental purposes without payment to me of any royalty thereon.

The present invention relates to an improved pressure exchanger characterized by an arrangement for producing both dynamic pressure exchange and gradual static pressure exchange.

The primary object of the invention is to provide a pressure exchanger of general utility in power cycles and gas turbine power plants capable of accomplishing a more efficient production of power by virtue of higher pressure ratios and greater operating flexibility.

A further object of the invention is to provide an improved pressure exchanger including at least one multicell rotor within a casing, low pressure and high pressure gas inlets on one side of the casing, low pressure and high pressure gas outlets on the other side of the casing to accomplish a dynamic pressure exchange in cells moving past said inlets and outlets, and conduit means making connection between separate cells intermediate between inlets and outlets to accomplish a gradual static pressure exchange between cells carrying gas at a higher pressure and cells carrying gas at a lower pressure.

Another object of the invention is to provide an improved pressure exchanger including at least one multicell rotor within a casing, low pressure and high pressure gas inlets on one side of the casing, low pressure and high pressure gas outlets on the other side of the casing with similar pressure passages substantially opposite to each other, and means to produce a gradual static pressure increase in cells moving from the low pressure gas inlet and outlet toward the high pressure gas inlet and outlet.

Another object of the invention is to provide an improved pressure exchanger including at least one multicell rotor within a casing, means to rotate the rotor, low pressure and high pressure gas inlets on one side of the casing, low pressure and high pressure gas outlets on the other side of the casing with similar pressure passages substantially opposite to each other, and conduit means extending between cells moving from the high pressure inlet and outlet toward the low pressure inlet and outlet and other cells moving from the low pressure inlet and outlet toward the high pressure inlet and outlet to effect a gradual static pressure exchange between cells carrying gas at a higher pressure and cells carrying gas at a lower pressure.

Another object of the invention is to provide an improved pressure exchanger having more than one multicell rotor with the rotors so correlated and interconnected by gas passages as to permit both a dynamic pressure exchange and a gradual static pressure exchange.

Another object of the invention is to provide a two-rotor pressure exchanger including means to rotate the rotors in opposite directions, low pressure and high pressure gas inlets at one side of each rotor, low pressure and high pressure gas outlets at the other side of each rotor

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with similar pressure passages substantially opposite to each other, and conduit means extending between cells of one rotor moving from the high pressure inlet and outlet toward the low pressure inlet and outlet and cells of the other rotor moving from the low pressure inlet and outlet toward the high pressure inlet and outlet to effect a gradual static pressure exchange between cells carrying gas at a higher pressure and cells carrying gas at a lower pressure.

The above and other objects of the invention will become apparent upon reading the following detailed description in conjunction with the accompanying drawings, in which:

Fig. 1 is a diagrammatic view of a power plant including a pressure exchanger to show one possible application of the invention.

Fig. 2 is a schematic cross sectional view of a simple form of pressure exchanger.

Fig. 3 is a diagrammatic developed plan view of the pressure exchanger of Fig. 2, which embodies only a dynamic pressure exchange.

Fig. 4 is a cross sectional view of one form of the invention for accomplishing both a dynamic pressure exchange and a gradual static pressure exchange.

Fig. 5 illustrates a modified construction for use in the apparatus of Fig. 4.

Fig. 6 is a transverse cross section taken on line 6—6 of Fig. 4.

Fig. 7 is a diagrammatic developed plan view to show the pressure exchange functions of the apparatus of Fig. 4.

Fig. 8 is a cross sectional view of a two-rotor pressure exchanger similar to that of Fig. 4 but with one rotor inside the other.

Fig. 9 is a diagrammatic developed plan view to show the pressure exchange functions of the apparatus of Fig. 8.

Fig. 10 is a fragmentary cross sectional view of a modified construction for use in the apparatus of Fig. 8.

Fig. 11 is an end view of a single rotor pressure exchanger capable of accomplishing both dynamic and gradual static pressure exchange.

Fig. 12 is a view of the other end of the single rotor pressure exchanger of Fig. 11, wherein a portion of the casing is cut away to show part of the rotor therein.

The pressure exchanger is a well-known type of power plant auxiliary and is especially adapted for use in power plants employing gas turbines as prime movers. By the addition of a pressure exchanger unit to such a power plant the overall efficiency can be increased very substantially. One prior disclosure of the pressure exchanger may be found in U. S. Patent No. 2,399,394 granted to Claude Seippel on April 30, 1946, and a later disclosure of a special power plant making use of a pressure exchanger may be found in U. S. Patent No. 2,461,186 granted to Claude Seippel on February 8, 1949. The possibilities for combining the pressure exchanger with various power plant units are many and varied, but by way of example one power plant or system is shown in Fig. 1 of the drawings. The main units as indicated therein are the air compressor, the fuel tank, the fuel pump, the combustion chamber, the pressure exchanger, the primary turbine, the secondary turbine and the power load which is a generator or dynamo in the example shown. The pressure exchanger is merely represented by a compressor **11** driven by a turbine **12** for the reason that if two such units were used in the system they would provide the closest known equivalent of a pressure exchanger. Thus the connected compressor and turbine **11** and **12** have been accepted as a functional representation of a pressure exchanger. This functional analogy is also carried through the theory and terminology

of pressure exchangers, especially by reference to the compressor cycle and the turbine cycle of the pressure exchanger. Starting with the air compressor receiving atmospheric air at 14.7 pounds per square inch, the incoming air enters at 1 and leaves at 2 having been boosted to a pressure of 50 pounds for example. The compressed air now enters the low pressure side of the compressor cycle (L_c) of the pressure exchanger. The high pressure side of the compressor cycle (H_c) connects at 3 with the combustion chamber, while the latter receives a steady supply of fuel by a conduit 4 from a fuel tank and fuel pump. The pressure boost through the compression cycle of the pressure exchanger may for example bring the air entering the combustion chamber up to about 150 pounds. The heated air leaving the combustion chamber has greatly expanded in volume over the volume of high pressure air which entered the combustion chamber and this heated air leaving at 5 supplies the secondary turbine by way of conduit 6 and also the high pressure side of the turbine cycle (H_t) of the pressure exchanger. The pressure of the air and combustion products reaching both of these units may be around 150 pounds for example. The exhaust from the secondary turbine flows in the conduit 7 and is combined with the outlet flow from the low pressure side of the turbine cycle (L_t) of the pressure exchanger, both at a pressure of 50 pounds for example. This combined gas flow now reaches the inlet of the primary turbine by way of the conduit 8. The exhaust from the primary turbine flows by way of a conduit 9 to any suitable unit or apparatus capable of utilizing the heat remaining in the exhaust gases. This exhaust pressure would of course be close to atmospheric pressure, that is, 14.7 pounds. As indicated at 10 a shaft connects the air compressor, primary turbine, secondary turbine, and the power load. Other arrangements may be effected if desired, for instance the secondary turbine may be used to drive the air compressor and some additional load, while the primary turbine may be used to drive the principal power load thus permitting different shaft speeds for the two turbines and the units driven thereby. This may be advantageous in securing closer speed regulation for the principal power load. As will appear in the subsequent description of the pressure exchanger, there is some power required to rotate the pressure exchanger but this is taken care of by a small electric motor or other auxiliary unit. It might be noted that the principal reason for the success of the pressure exchanger is that it does provide the equivalent of a coupled compressor and turbine but does so with less complicated machinery, at less cost and with less space requirement. Also the maintenance problems are far less significant with the pressure exchanger.

The symbolic representation of the pressure exchanger Fig. 1 may be noted now in particular. The compressor 11 and coupled turbine 12 are shown as though they are driven by a motor 13, but it is understood that the compressor and turbine are merely shown because of their theoretical equivalency to a pressure exchanger. It should be remembered that the cold gases flowing through the compressor cycle from L_c to H_c are being compressed, while the hot gases flowing through the turbine cycle from H_t to L_t are being expanded. Thus the turbine cycle may also be termed the expansion cycle. One characteristic of the pressure exchanger is that the gas volumes of the compressor cycle and the turbine cycle must be equal, because they are handled in the same rotor cells making up the pressure exchanger rotor or rotors. This fact explains why it is necessary to provide a bypass flow around the turbine or expansion cycle, as at 6 in the power plant of Fig. 1. A simple form of pressure exchanger is shown in longitudinal cross section in Fig. 2, wherein there is a cylindrical housing 14 having a rotor 15 mounted to spin therein on the shaft 16 connected to the driving motor 17. On the outer periphery of the rotor 15 there are numerous axially disposed cells 15′ open at both ends so as to connect in successive manner to the conduits L_c, L_t, H_t and H_c. The air in conduit L_c and also that filling about one-half of one cell 15′ represents air to be compressed, while the air in conduit H_c and also filling about one-half of another cell 15′ represents air which has been compressed. Similarly the air or combustion products in conduit H_c represents gas to be expanded, while the gas in conduit L_t represents gas which has been expanded.

Briefly the process accomplished in the pressure exchanger of Fig. 2 takes advantage of dynamic exchange by means of shock waves, whereby air trapped suddenly in a cell is compressed by a ram effect induced by the flow of low pressure air and other gas is later compressed by the action of a second shock wave set up in connecting a cell with a source of high pressure gas. This alternate trapping and compressing of air takes place in the lower and upper cells or channels 15′ respectively, in the apparatus shown in Fig. 2. The low pressure air flows in a selected cell at L_c, while the same air after compression flows out of the cell at H_c. The first compression is accomplished by suddenly stopping the air moving along the conduit L_c, since the cell receiving a charge of air immediately moves to a position where there is no outlet connected thereto. The second compressive shock wave set up by suddenly connecting the cell to the conduit H_t not only causes additional compression of the air previously trapped in the cell but also forces the high pressure gas into the conduit H_c. This may be considered an expansion wave, the action of which is duplicated when the cell again becomes connected to the conduit L_t. Here as the air flows into the cell from the conduit L_c the gas which was trapped in the cell when it was connected to the conduit H_t will flow out by expansion into the conduit L_t. It must be noted too that the rotor 15 is turning on its axis at a rapid rate of speed, so that the successive steps of the process as applied to a selected rotor cell occur with very little time lag. While the rotative speed may vary within reasonable limits even for the same unit, a round figure for the linear speed of the cells may be stated by way of example as 400 feet per second. The difference in temperature of the incoming air or gas at conduits L_c and H_t causes the cell structure to reach only a mean temperature, since the air or gas is in the cells for only a brief interval of time. Furthermore the whole rotor structure becomes uniformly heated soon after operation begins and there will be no appreciable heating and cooling of the cell walls at various points on the cycle of operation. For a more detailed description of the operation of the basic dynamic pressure exchanger, reference may be had to the Seippel Patent No. 2,399,394 cited above.

For a description of the preferred embodiment of the present pressure exchanger which provides for both static and dynamic pressure exchange, reference is now made to Fig. 4 wherein there is illustrated a two-rotor pressure exchanger with counter-rotating rotors 20 and 21 of identical size and construction. The rotor 20 is directly connected to a motor 22 and the rotor 21 is directly connected to a motor 23. The motors are of the synchronous type and are supplied from the same source of alternating current so as to run at exactly the same speed but in opposite directions of rotation. As an alternative arrangement a motor may be directly connected to one rotor only and the other rotor may be driven in unison with the first rotor by the use of a set of bevel gears as indicated at 24 in Fig. 5. Thus this alternative construction will positively maintain the same rotor speed for the counter-rotating rotors and may even be employed where separate motors are provided, as in Fig. 4, to ensure perfect synchronization of the rotors. Each of the rotors 20 and 21 carry individual cells or

channels arranged around the periphery thereof, as shown best in Fig. 6. It is understood that these cells are open at each end to permit gas flow therethrough when the cells coincide with the inlet and outlet conduits. The rotor **20** is surrounded by a cylindrical casing member **26**, while the rotor **21** is surrounded by a similar casing member **27**. The remote ends of these members **26** and **27** are closed by end plates **28** and **29** respectively. The adjacent ends of the casing members are closed by additional end plates **30** and **31** which are provided with central bearing bosses **30'** and **31'** secured together as shown. Between the end plates **30** and **31** are a pair of opposite gas inlet conduits **32** and **33**. These conduits are provided with side openings coinciding with other openings in the casing end plates and these openings are of such size and shape as to open into more than one rotor cell at a time, as will be explained with reference to Fig. 7. Fixed to the casing end plates **28** and **29** are pairs of gas outlet conduits **34, 34'** and **35, 35'**. Between the adjacent casing end plates **30** and **31** there are static pressure exchange conduits or passages, which merely extend between the adjacent ends of the rotor cells. While these passages do not show in Fig. 4 they are shown in Fig. 7, wherein the rotor cells and various conduits are shown in a developed diagrammatic plan view.

Before considering Fig. 7 of the operation of the basic dynamic pressure exchanger of Fig. 2 will be explained by reference to Fig. 3, which shows in developed plan the cells of rotor **15** and the various conduits L_c, L_t, H_t and H_c. There is a duplication of conduits L_c and L_t so that the operation may be explained with reference to more than one complete revolution of the rotor. The cells are of course assumed to be moving in the direction of the arrow A. The letter L indicates low pressure, H indicates high pressure and the subscripts $_c$ and $_t$ indicate the compressor cycle and the turbine cycle respectively. The low pressure air flowing in at L_c is at a lower temperature than the high pressure gas flowing in at H_t, since by reference to the power plant diagram (Fig. 1) it may be observed that the low pressure air supply is merely derived from the air compressor while the high pressure gas supply is derived from the combustion chamber. The low pressure gas or air which is to be compressed further flows into the pressure exchanger at L_c displacing the expanded hot gases which are simultaneously flowing out of the pressure exchanger at L_t. This is termed the low pressure scavenging phase and the direction and magnitude of the scavenging velocity in the inlet conduit L_c is so selected that the net velocity component relative to the rapidly moving cells is approximaely parallel to the cell walls. Thus in a cell arrangement as shown in Fig. 3 the resultant net velocity of the scavenging flow will also be parallel to the rotor axis, but it is understood that in some designs the cell walls would necessarily vary from the true axial alignment as illustrated. If the design is properly worked out the scavenging flow will not impinge on the cell walls but will travel through the cells in a direction parallel to the cell walls, that is the direction of flow will not be changed by the rotor. Thus the zone of contact between the scavenging air flowing in at L_c and the expanded hot gasses flowing out at L_t travels through the rotor with unchanged direction as long as all conditions of operation are at a steady state. This zone will be marked by the line **2—3** of Fig. 3 and will not vary in position to an appreciable extent during the operation of the pressure exchanger. As each cell passes the point **3** the velocity of the air is suddenly brought to zero and the shock effect produces a compression wave traveling back along the line **3—4**. As this shock wave reaches the point **4** the cell is closed and the result is that air is trapped in the cell at increased pressure and is carried on as the cell moves toward the point **6**.

Now considering the high pressure scavenging phase which commences when the cells reach point **6**, it will be seen that high pressure gas flowing in conduit H_t will generate a compression wave to further compress the air which was trapped in the cells. At the same time the air will be caused to move across the cell with a scavenging velocity which will be in the general range of the velocity of the gas flowing in conduit H_t. The zone of contact between the trapped air and the incoming high pressure gas will extend along the line **6—7** but the compression wave takes a path along the line **6—5**. The air ahead of the compression wave is further compressed and behind the wave there is a velocity of gas flow which determines the velocity of flow into the outlet conduit H_c. As the cell reaches the point **8** the flow from the conduit H_t into the cell is stopped but the zone of contact is still moving across the cell. Thus a first rarefaction wave originates at point **8** producing an expansion of air accordingly. This expansion wave flowing in the direction of line **8—7** intersects the zone of contact **6—7** at the cut-off point **7**, with the result that the cell will move on containing gas at an intermediate pressure and temperature. As the cell reaches point **1** the gas starts to flow out into conduit L_t where the pressure is relatively low. Thus a second expansion wave starts to propagate in the cell and produces a flow of gas at a velocity dependent on the pressure difference behind and ahead of the expansion wave. This expansion wave flows in the direction **1—2** and as the wave reaches the point **2** the cell is opened to the conduit L_c, permitting fresh air to flow into the cell for the low pressure scavenging phase first described.

The fundamental or basic dynamic pressure exchanger as described with relation to Figs. 2 and 3 operates by means of two compression waves and two expansion waves. The optimum pressure ratios are obtained by careful selection of scavenging velocities for the low pressure and high pressure scavenging phases and by carefully determining the geometrical correlation between the points **1** to **8** in Fig. 3. Fans in the conduits are also used in most installations to influence the scavenging velocities.

The present invention seeks to improve the action of the basic dynamic pressure exchanger by the use of a combined static and dynamic pressure exchange and by so combining these two effects a machine results which will have improved characteristics impossible of attainment by applying either of these principles alone. The pressure exchanger so constructed will achieve a higher pressure ratio and will maintain high efficiency over a wider range of variables, such as temperature and pressure conditions in the inlet and outlet conduits and rates of gas flow.

In Fig. 7 there is shown a development of the periphery or preipheral portions of a dual rotor pressure exchanger, the structure of which is shown in cross section in Fig. 4. The two rotors **20** and **21** rotate in opposite directions as indicated by the arrows A and A'. The outlet conduits L_t, H_c and L_t', H_c' correspond with the conduits **34, 34'** and **35, 35'** of Fig. 4. The inlet passages L_c and H_t between the rotors correspond with conduits **32** and **33** of Fig. 4, while the static pressure passages not seen in Fig. 4 are indicated at **36** in Fig. 7. It is of course understood that the outlet conduits L_t and L_t' will be connected to a common header, as will also the outlet conduits H_c and H_c'. In some power plants the outlet conduits may be connected separately to units of their own. For instance in a power plant built according to the principles of Fig. 1 the conduits H_c and H_c' may extend directly to separate combustion chambers and the conduits L_t and L_t' may extend to separate gas turbines.

To explain the operation of the dual rotor machine with combined static and dynamic pressure exchange only a single cell of rotor **20** and a single cell of rotor **21** will be considered and the cell action will be traced

always in the direction of its peripheral movement (arrow A and arrow A'). Starting at left-hand end of rotor 20 the cell is filled with hot gas at a moderate pressure and on reaching conduit L_t flows out under the influence of an expansion wave which is set up by the pressure difference between the air pressure in the cell and in the conduit L_t. As the expansion wave is under way the cell reaches the conduit L_c and the air therefrom flows into the cell to start the low pressure scavenging phase. The zone of contact between the low pressure scavenging air and the outflowing warm air will be marked by the line 2—3. As the cell reaches the cut-off point 3 the low pressure compression wave is started due to the sudden closing of the cell. This will cause a compression of the air in the cell, the compression wave traveling in the direction 3—4 and reaching the end of the cell coincident with the cut-off point 4. Now air at a moderate pressure is trapped in the cell and carried on at the peripheral speed of the rotor. Now as the cell reaches the first static pressure passage or distributing passage 36 past the point 4 it is momentarily connected to a supply of high temperature air or gas at a higher pressure than the air in the cell, the supply being obtained from cells of the other rotor 21. Thus a small amount of hot gas from rotor 21 is delivered to the cell of rotor 20 further compressing the air therein. Thus compression continues in increments as other passages 36 are reached, and a gradual static pressure exchange is effected to build up the air pressure in the selected cell of rotor 20 before reaching the inlet conduit H_t, where the second compression wave and the high pressure scavenging phase originate.

As the selected cell of rotor 20 reaches the point 6 it is connected with the inlet conduit H_t filled with high temperature gas at high pressure. This gas starts a compression shock wave in the direction 6—5, with the zone of contact between the high pressure gas and lower pressure gas extending along the line 9—7. This compression wave completes the high pressure scavenging phase and by its compressive force brings the cell contents to the final high pressure. This compressed air flows out through the conduit H_c as the cell becomes substantially full of hot gases flowing in at H_t. This gas being at a relatively high pressure, it now flows through the successive passages 36 toward the cells of rotor 21 after the cell passes the cut-off point 7 and this action continues until the cell reaches the point 1 where the gas begins the expansion into conduit L_t.

Considering now the rotor 21 it will be seen that a selected cell beginning at the right-hand side of Fig. 7 is going through the low pressure scavenging phase following expansion of the gas into conduit L_t'. The compression wave induced by moving the cell past the cut-off point 3' compresses the fresh air flowing in from inlet conduit L_c and this air or gas at increased pressure is trapped in the cell as the cell passes the cut-off or trapping point 4'. The cell is now filled with air at a moderate pressure and soon reaches the first static exchange passage 36 to the left of point 4'. The high temperature, high pressure gas from cells of rotor 20 now located between points 1 and 7 effect a gradual static pressure boost by their flow through the passages 36 in the direction of the arrows and as a result the pressure in the selected cell of rotor 21 builds up as it progresses from point 4' to point 6'. As the cell reaches the point 6' the contents thereof become subject to the full pressure in the conduit H_t and a compression wave moves in the direction of line 6'—5', to further boost the pressure of the cell contents, which now flow out through conduit H_c'. The high temperature, high pressure air trapped in the cell as it passes point 7' will now be capable of effecting a static pressure boost in cells of rotor 20 by reason of the pressure distribution passages 36. At the same time there is little loss of pressure in the cell before it reaches the outlet conduit L_t' where the gas can expand and flow into the conduit L_t'. After and during expansion the low pressure scavenging phase takes place under the action of the fresh air flowing in at L_c.

In a pressure exchanger operating in the manner above outlined it will be possible to achieve greater pressure ratios and a more efficient power plant can thus be built, and such a plant will be less affected by load variations and other transient changes. As should be clear from the description of the pressure interchange through the distributing passages 36, each rotor acts on the other in a unique and highly satisfactory manner. As the high pressure compressive waves begin in the cells of rotors 20 and 21 at the points 6 and 6', the cell pressure is already built up by static pressure exchange and the pressure in the conduits H_c and H_c' will be higher than would be the case otherwise.

As indicated above the dual rotor pressure exchanger may be used in a power plant as shown in Fig. 1 or in any similar power cycle. As will be appreciated, the increased pressure boost from L_c to H_c will give a better output from the combustion chamber thus furnishing more power at the secondary turbine and greater pressure at the pressure exchanger inlet conduit H_t. Thus the power plant efficiency will increase very decidedly and the pressure exchanger itself will be less sensitive to variable factors. This flexibility should be especially advantageous in locomotive power plants where there may be variations in load, barometric pressure and air temperature. The same would apply also in ship, aircraft and stationary power plants. The increasing use of gas turbine power plants make it desirable to improve any adjuncts or auxiliaries used in association therewith.

The dual rotor pressure exchanger with combined static and dynamic pressure exchange may take other forms than that described above. For example one rotor may run inside the other as illustrated in Fig. 8. Considering the structure of Fig. 8 in detail it will be noted that the inner rotor 40 is carried on a central shaft 40' while the outer rotor 41 is mounted to turn on an axial extension 42 of the inner rotor 40. The shaft 40' is rotatably mounted in a bearing 43 at one end and is connected to a motor 44 at the other end. The inner rotor 40 includes an axial extension 42' which is non-rotatably secured to shaft 40' by means of a crosspin 43'. Surrounding the two rotors is drum-like casing or housing 45 having a cylindrical inner wall 46 extending between the outer periphery of the rotor 40 and the inner periphery of the rotor 41. This wall 46 is provided with static pressure exchange openings 47, corresponding in function with the passages 36 of Fig. 7. These exchange ports or openings 47 may extend over the whole periphery of the machine or may cover only certain portions or segments of the periphery as will be better described below in connection with Fig. 9. The inner rotor 40 runs at the same speed as the motor 44 and in the same direction, while the outer rotor 41 is made to operate at the same speed as the motor but in the opposite direction. This is accomplished by the use of a bevel gear 48 on the rotor extension 42 which drives the intermediate bevel gears 49. The gears 49 in turn mesh with a bevel gear 50 fixed to the axial extension 51 of the rotor 41. The opposite bevel gears 49 are provided with stub shafts mounted to turn freely in the fixed bearing 49'.

Connected at the left-hand side of casing 45 are the gas inlet conduits 52 and 53 and at the right-hand side of the casing are gas outlet conduits 54 and 55. These conduits correspond to those designated H_t, L_c, H_c and L_t respectively in Fig. 7. The inner rotor 40 carries cell partitions on its outer periphery like the rotor illustrated in Fig. 6. The rotor 41 is built similar but is hollow to receive the counter-rotating rotor 40. The radially extending wall of the rotor 41 is provided with successive openings 56 all the way around its periphery to allow passage of gas from the cells of rotor 40 out through the conduits 54 and 55. Also the axially extending wall of the rotor 41

is provided with successive openings 57 which allow the static pressure exchange through the openings 47 described above.

The operation of the pressure exchanger made according to Fig. 8 is similar to that of Fig. 4 except for the static pressure exchange which follows a short and direct route through ports 47 and 57. Fig. 9 shows a functional diagram similar to Fig. 7 so that the operation may be visualized by showing a development plan of the two rotors, one above the other. As indicated by the arrows B and B′ the two rotors rotate in opposite directions. The compression and expansion wave flow is exactly the same as in Fig. 7, except that the gradual static pressure exchange instead of being obtained by a flow through the conduits or passages 36 in Fig. 7 is now accomplished by a radial flow through the ports 47 and 57 in the dividing wall 46 and in the cylindrical wall portion of rotor 41, respectively. The ports 57 may extend clear around the rotor wall with just enough wall material between each port to hold the structure rigidly together, but the ports 47 in the dividing wall may have an extent as indicated in Fig. 9. Instead of each port 47 being continuous over a substantial section of the periphery, as indicated in Fig. 9, these ports may be divided up into a group of separate passages similar to the spaced passages 36 of Fig. 7. As will be observed, the development plan view of Fig. 9 covers more than one complete revolution of the rotors just as in Fig. 7. As will be appreciated by comparing Figs. 7 and 9, the static pressure exchange is always from a group of cells at high pressure, or in compression, to a group of cells at low pressure, or in expansion. This will mean that flow through one port 47 will be in a direction opposite to flow through the adjacent port 47. The result is the partial compression areas bounded on one side by the lines 4, 9 and 2′, 9′. Thus in Fig. 9 flow through the left-hand port 47 will be from the lower rotor moving in direction B′ to the upper rotor moving in direction B. Flow through the right-hand port 47 will be from the upper rotor moving in direction B to the lower rotor moving in direction B′. While there will only be two ports 47 in the apparatus of Fig. 8 and Fig. 9, they are shown superimposed on both rotors in the diagrammatic plan in order to show the relative position with respect to each rotor and with respect to the gas inlet and outlet passages. The arrangement of rotors as shown in Fig. 8 offers the advantage of providing the shortest possible static pressure exchange conduits and thus minimizes pressure and heat loss, as well as affording a more compact machine.

In Fig. 10 there is illustrated a portion of a two-rotor pressure exchanger similar in principle to that of Fig. 8. The section shown is between the gas inlet passages 52 and 53 and shows an alternative arrangement of the static pressure exchange passages. The rotor casing is made up of end walls 60 and 61 and outer cylindrical wall 62. Mounted to rotate therein are the rotors 63 and 64 having the usual cells on their outer periphery. The axially extending wall 65 of the rotor 64 is not ported as in Fig. 8, but instead the end wall 60 is provided with channel 66 to form a pressure exchange passage 67 corresponding in relative peripheral position and operating effect to the port 47 of Fig. 8. This arrangement provides for purely axial flow of the exchange pressure and avoids cross currents which will tend to mix two working gases at different temperatures. For this reason the modified arrangement of static pressure exchange ports may prove more practical than the construction shown in Fig. 8.

As will be understood from the foregoing description the two rotors can be arranged in several ways and may even be completely independent as far as their housings and bearing supports are concerned. However in all cases the inlet and outlet conduits must be arranged to achieve dynamic pressure exchange and the static pressure exchange passages must be provided to achieve a gradual static pressure exchange, in order to increase the efficiency of the machine over that of previously known pressure exchangers.

The main purpose of the present invention is to obtain static and dynamic pressure exchange. While the preferred form of the invention employs two rotors operating in opposite directions of rotation, it is possible to achieve the desired results by the use of a single rotor. Such a single rotor pressure exchanger is shown in Figs. 11 and 12. The general arrangement of the rotor and housing is like the basic pressure exchanger of Figs. 2 and 3. Figs. 11 and 12 show the improved single rotor pressure exchanger as seen from opposite ends. The cylindrical casing will thus be closed at its ends by circular end plates 68 and 69, and the end plates will each be provided with rotor shaft bearings 70. The rotor 71 mounted within the casing or housing is provided with peripheral cells 72 formed by providing radial partitions 73 on the outer rim of the rotor. The end plate 68 is provided with gas inlet conduit 74 corresponding to conduit H_t of Figs. 2 and 3, and also with gas inlet conduit 75 corresponding to conduit L_c of Figs. 2 and 3. Between these gas inlets there are arranged static pressure exchange conduits 76, 77, 78, 79 and 80 through which static pressure exchange occurs in the direction of the arrows adjacent to the conduits. The other end plate 69 is provided with gas outlet conduit 84 corresponding to conduit H_c of Figs. 2 and 3, and also with gas outlet conduit 85 corresponding to conduit L_t of Figs. 2 and 3. The direction of rotation of the rotor 71 should be in the direction of counterclockwise arrow A in Fig. 12. However the side of the rotor facing outward in Fig. 11 will be turning in a clockwise direction. Looking at Fig. 11 the cells 72 will be moving from the conduit H_t toward conduit L_c as they pass the inlet ends of static pressure exchange conduits 76 to 80 inclusive. The pressure exchange obtained will correspond to that depicted in Fig. 7, since referring to the latter view it will be seen that the rotor cells which are moving from inlet conduit H_t toward inlet conduit L_c are connected to the inlet end of conduits 36, while the cells which are moving from inlet conduit L_c to the inlet conduit H_t are connected to the outlet end of conduits 36. In the same way in Fig. 11 the cells 72 moving from conduit H_t toward L_c are connected to the inlet ends of static pressure exchange conduits 76 to 80, while the outlet ends of the exchange conduits at the left of Fig. 11 will be connected to other cells of the same rotor moving from conduit L_c toward conduit H_t. Thus the analogy between the two-rotor exchanger and the single rotor exchanger is perfectly clear, although the relatively long static pressure exchange conduits 76 to 80 present one disadvantage in the single rotor machine. For smaller power plants however the single rotor exchanger will have a distinct advantage over the two-rotor exchanger.

As now understood the previously known pressure exchanger operate either on the principle of a simple static pressure exchange or a gradual static pressure exchange or a dynamic pressure exchange. The present invention combines a gradual static pressure exchange with a dynamic pressure exchange, thus producing results which can not be obtained by the use of either effect alone. The simple static pressure exchanger has the disadvantage that the scavenging velocities must be kept low in order to avoid excessive losses. The capacity of these machines is therefore limited by the low velocities. The dynamic pressure exchanger on the other hand can in principle apply much higher scavenging velocities but has other disadvantages. The basic dynamic pressure exchanger as shown in Figs. 2 and 3 operates by means of two compression and two expansion waves. The relations between the scavenging velocities and the pressure ratios for each of these waves may be determined by an equation for dynamic relation of the gas flow.

The four resulting gas-dynamic conditions work together in the operation of the machine and must be satisfied in addition to static pressure and energy conditions. This means that there are limiting conditions influencing design and operation, which conditions determine the feasible pressure ratios, the efficiency and also the machine capacity.

The present invention avoids the disadvantages of the previously known pressure exchangers to a large extent. By a combined static pressure and dynamic pressure exchange the total pressure ratio can be made within certain limits independent of the scavenging velocities. The total achievable pressure ratio of the new machine using the present principles of invention is composed of a static and a dynamic component:

$$R_{total} = P_h/P_1 \doteq R_{static} \times R_{dynamic}$$

The dynamic component in this equation depends of course on the scavenging velocities but the relations concerned are simpler and clearer because the four conditions in the exchanger corresponding to the four waves are now more or less isolated. The static pressure component R_{static} is nearly independent of the scavenging velocities and depends mainly on the ratio of the weight flow through the turbine and compressor cycles and on the temperature conditions. The static pressure ratio can be made even smaller than unit, which means that in the middle stages of static pressure exchange the pressure of the fresh air is actually higher than that of the hot combustion gases. With such a condition the equalizing flow in the exchange conduits would have therefore the opposite direction than that shown in the drawings and these conduits would be filled with fresh air instead of hot combustion gases. This shows the extreme flexibility of operation of the present apparatus when conditions are changing. However it should be understood that under average working conditions the static pressure ratio will be greater than unity and the static pressure exchange flow will be in the directions specifically marked on the drawings or explained above.

The embodiments of the invention herein shown and described are to be regarded as illustrative only and it is to be understood that the invention is susceptible of variations, modifications and changes within the scope of the appended claims.

I claim:

1. A pressure exchanger comprising two multi-cell rotors each of which carries a plurality of similar cells on its periphery extending from one side to the other of the rotor with each cell being separated from adjacent cells by radially extending walls, means to enclose said rotors, means to rotate said rotors on their central axes in reverse relation, means providing a first gas inlet channel at one side of said rotors for supplying gas at low pressure to the rotor cells, means providing a second gas inlet channel at said one side of said rotors and circumferentially spaced from said first gas inlet channel for supplying gas at a high rate pressure to the rotor cells, means providing a first and a second gas outlet channel at the other side of said rotors for delivering from said cells the low pressure gas having its pressure increased to the value of the high pressure gas and for delivering from said cells the high pressure gas having its pressure reduced to the inlet pressure of the low pressure gas, said inlet and outlet channels being positioned circumferentially so that the low pressure outlet is reached by the rotor cells before the low pressure inlet and the low pressure outlet is passed by the rotor cells before the low pressure inlet and wherein the time intervals involved are determined by the time required for an expansion wave and a compression wave respectively to traverse said cells, and so that the high pressure inlet is reached by the rotor cells before the high pressure outlet and the high pressure inlet is passed by the rotor cells before the high pressure outlet and wherein the time intervals involved are determined by the time required for a compression wave and an expansion wave respectively to traverse said cells, and means providing static pressure exchange passages between the cells of the respective rotors to connect cells of one rotor moving from high pressure inlet closing toward low pressure inlet opening with cells of the other rotor moving from low pressure inlet closing toward high pressure inlet opening and which also connect cells of said one rotor moving from low pressure outlet closing toward high pressure outlet opening with cells of the other rotor moving from high pressure inlet closing toward low pressure inlet opening, the ends of said static pressure exchange passages connecting with said cells at the axial ends thereof and having a radial extent at the ends equal to the radial extent of said cells, whereby the pressure in connected cells is gradually equalized and some expanding gas is delivered from cells at high pressure to cells at low pressure.

2. A pressure exchanger comprising two multi-cell rotors each of which carries a plurality of similar cells on its periphery extending from one side to the other of the rotor, means to enclose said rotors, means mounting said rotors for rotation on their central axes in adjacent coaxial relation, means to rotate said rotors in reverse relation, means providing a first gas inlet channel at the adjacent sides of said rotors for supplying gas at low pressure to the rotor cells, means providing a second gas inlet channel at the adjacent sides of said rotors and circumferentially spaced from said first gas inlet channel for supplying gas at a high pressure to the rotor cells, means providing a first and a second gas outlet channel at the remote sides of said rotors for delivering from said cells the low pressure gas having its pressure increased to the value of the high pressure gas and for delivering from said cells the high pressure gas having its pressure reduced to the inlet pressure of the low pressure gas, said inlet and outlet channels being positioned circumferentially so that the low pressure outlet is reached by the rotor cells before the low pressure inlet and the low pressure outlet is passed by the rotor cells before the low pressure inlet and wherein the time intervals involved are determined by the time required for an expansion wave and a compression wave respectively to traverse said cells, and so that the high pressure inlet is reached by the rotor cells before the high pressure outlet and the high pressure inlet is passed by the rotor cells before the high pressure outlet and wherein the time intervals involved are determined by the time required for a compression wave and an expansion wave respectively to traverse said cells, and means providing static pressure exchange passages between the adjacent sides of the rotors to connect cells of one rotor moving from high pressure inlet closing toward low pressure inlet opening with cells of the other rotor moving from low pressure inlet closing toward high pressure inlet opening and which also connect cells of said one rotor moving from low pressure outlet closing toward high pressure outlet opening with cells of the other rotor moving from high pressure inlet closing toward low pressure inlet opening, whereby the pressure in connected cells is gradually equalized and some expanding gas is delivered from cells at high pressure to cells at low pressure.

3. A pressure exchanger comprising two multi-cell rotors each of which carries a plurality of similar cells on its periphery extending from one side to the other of the rotor, one of said rotors being made hollow to receive the other and smaller rotor in enclosing relation therein, means providing a housing around said rotors, means mounting said rotors for independent rotation on their central axes, means to rotate said rotors in reverse relation, means providing a first gas inlet channel at the corresponding sides of said rotors for supplying gas at low pressure to the rotor cells, means providing a second gas inlet channel at said corresponding sides of said rotors and circumferentially spaced from said first gas inlet channel for supplying gas at a high pressure to the rotor cells, means providing a first and a second gas outlet channel

at the other corresponding sides of said rotors for delivering from said cells the low pressure gas having its pressure increased o the value of the high pressure gas and for delivering om said cells the high pressure gas having its pressure r ced to the inlet pressure of the low pressure gas, sai ilet and outlet channels being positioned circumferentially so that the low pressure outlet is reached by the rotor cells before the low pressure inlet and the low pressure outlet is passed by the rotor cells before the low pressure inlet and wherein the time intervals involved are determined by the time required for an expansion wave and a compression wave respectively to traverse said cells, and so that the high pressure inlet is reached by the rotor cells before the high pressure outlet and the high pressure inlet is passed by the rotor cells before the high pressure outlet and wherein the time intervals involved are determined by the time required for a compression wave and an expansion wave respectively to traverse said cells, and means providing static pressure exchange passages between said corresponding sides of said rotors to connect cells of one rotor moving from high pressure inlet closing toward low pressure inlet opening with cells of the other rotor moving from low pressure inlet closing toward high pressure inlet opening and which also connect cells of said one rotor moving from low pressure outlet closing toward high pressure outlet opening with cells of the other rotor moving from high pressure inlet closing toward low pressure inlet opening, whereby the pressure in connected cells is gradually equalized and some expanding gas is delivered from cells at high pressure to cells at low pressure.

4. A pressure exchanger as defined in claim 3 wherein said means to rotate said rotors in reverse relation comprises an axial extension on each rotor with one extension extending inside the other, a bevel gear wheel on the outer end of each extension with the beveled sides of the respective wheels facing each other, bevel gears between said wheels and in constant mesh therewith, and a motor directly connected to one of said extensions.

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JET REACTION PROPULSION UNITS UTILIZING A PRESSURE EXCHANGER

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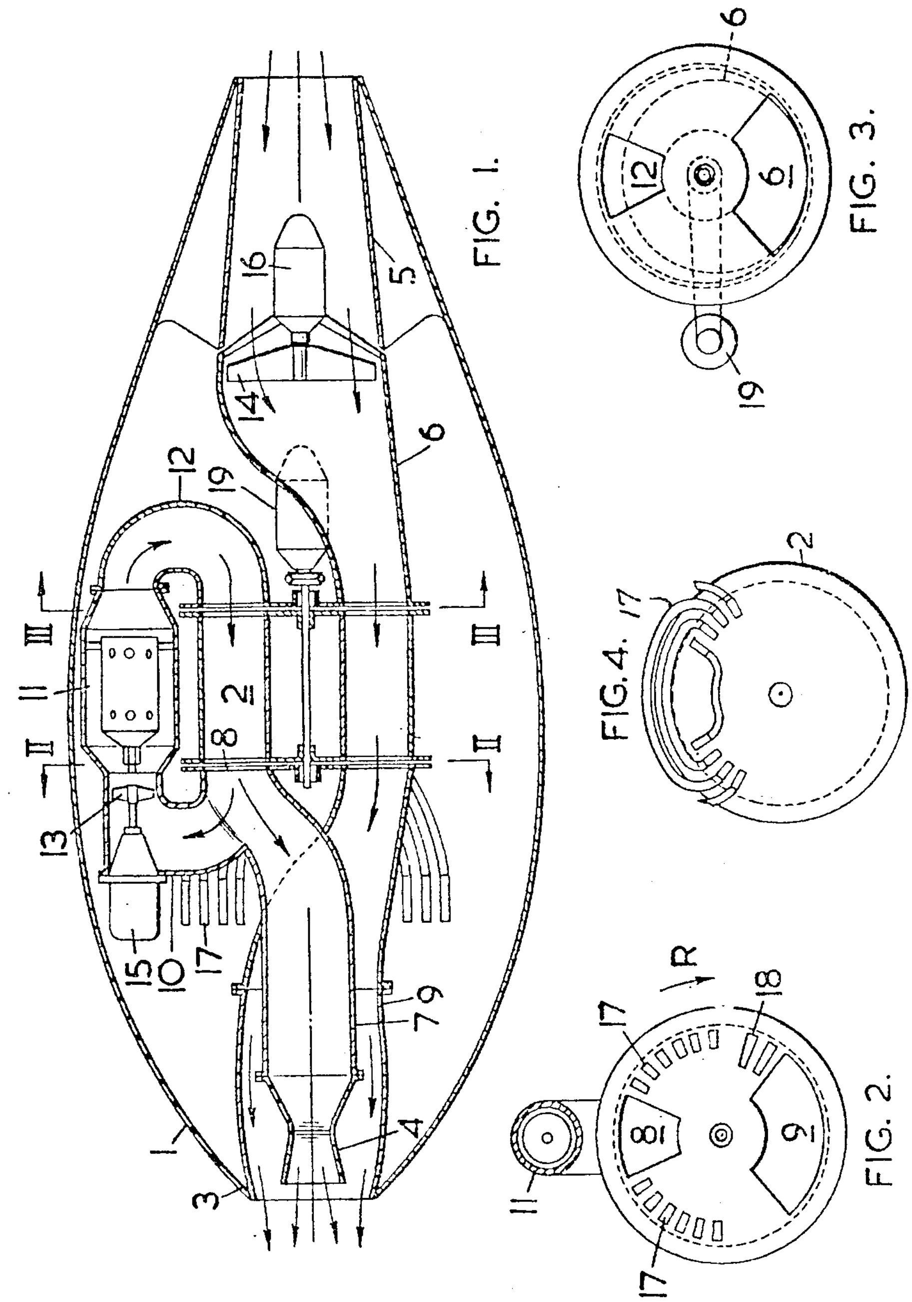

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JET REACTION PROPULSION UNITS UTILIZING A PRESSURE EXCHANGER

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FIG.5.

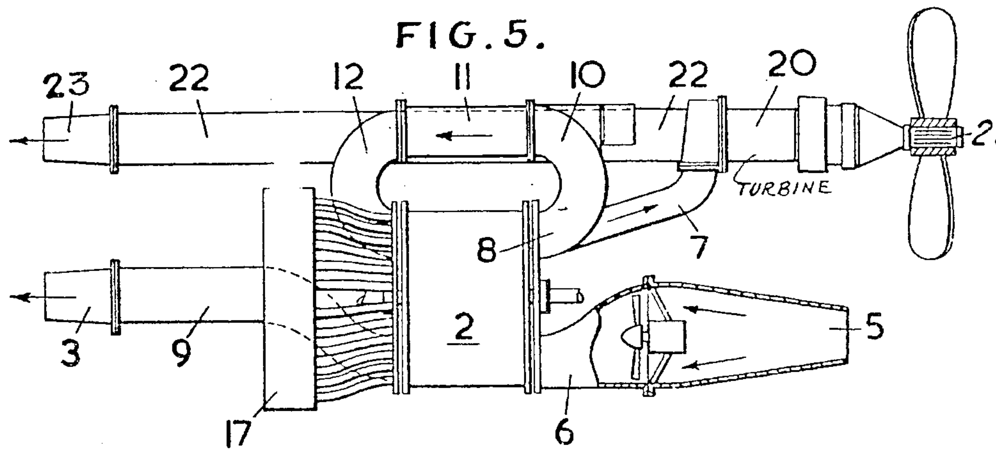
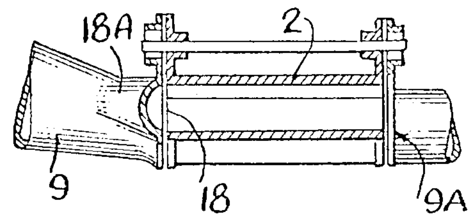


FIG.2A.



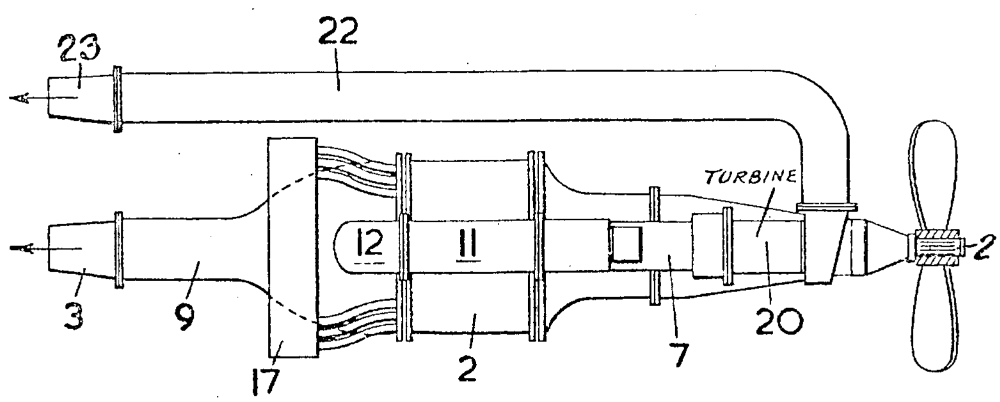


FIG. 6.

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United States Patent Office

2,757,509

Patented Aug. 7, 1956

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2,757,509

JET REACTION PROPULSION UNITS UTILIZING A PRESSURE EXCHANGER

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Application June 5, 1950, Serial No. 166,283

Claims priority, application Great Britain June 14, 1949

11 Claims. (Cl. 60—35.6)

This invention relates to the propulsion of vehicles, by which is herein meant more particularly aircraft (including aerial missiles), but also such marine craft and land vehicles as are amenable to jet propulsion, either as a main or an auxiliary source of power.

The invention deals mainly with the application to reaction propulsion of the devices known as pressures exchangers by which term is to be understood rotary machines (being heat engines working with gaseous fluid), which comprise at least one rotor embodying cells arranged as a circular series (or the alternative mentioned in the next following paragraph), the working cycle of which machines involves the compression of gas in some cells of the series and the simultaneous expansion of gas in other cells of the series, the compression and expansion stages thus formed being associated with heat input and heat rejection stages (at high or low pressure) involving the flow of gas into, and/or out of, the cells.

The above definition of pressure exchangers will be assumed to include the possible case in which a circular series of cells is accommodated in non-rotary structure and co-operates with rotary structure embodying ducts for the gas flow associated with the heat input and rejection stages.

The present invention provides apparatus for vehicle propulsion comprising a pressure exchanger, constructed to function by taking in air and delivering a supply of high pressure hot gas, and which is scavenged at least at the heat rejection stage, nozzle means for the expansion of a gaseous stream to form a propulsive jet, and ducting for supplying, from the pressure exchanger to the nozzle means, gas which is at a suitable temperature and pressure to provide by expansion through the nozzle means, useful propulsive thrust which either constitutes, or assists, the main means of propelling the vehicle.

If the main output of the apparatus is in the form of thrust supplied by the nozzle means, then the latter may be supplied with gas rejected by the pressure exchanger during the heat input stage, and possibly, to provide further thrust. Gas removed during the low pressure scavenging associated with heat rejection, may also be expanded through nozzle means to provide additional thrust. If the main output is in the form of mechanical work provided by a heat engine (e. g. a turbine) in which is expanded the gas rejected by the pressure exchanger at heat input, then nozzle means (which in this case furnishes auxiliary propulsive effort only) may be supplied only with gas removed from the pressure exchanger during the low pressure scavenging stage. Thus the said heat engine (e. g. turbine) may drive a normal airscrew, which is merely assisted by thrust from the nozzle means.

Preferably, in either case, the pressure exchanger is also scavenged at the heat input stage.

To augment the thrust in either case, each cell about to undergo the low pressure scavenging associated with heat rejection may be placed into communication, before being scavenged, with expansion nozzle means so that the gas in such cell is allowed to expand before reaching the low pressure scavenging zone, thereby furnishing propulsive thrust.

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Pressure exchangers in which all or part of the pressure rise and fall occurring in the machine is effected by waves of compression and expansion may usefully be used in the present invention.

The so-called "exchange of pressure" between gas which is expanding and gas which is being compressed (e. g. between expansion cells which are in communication by "transfer gas" apertures with compression cells), which gives rise to the name of the machine under discussion, will, generally speaking, result at most in the equalization of the pressure difference between the communicating zones. It may be possible however, by appropriate designing, to further continue the exchanging process beyond the equalization point i. e. until the pressure of the gas which has been compressed has risen above that of the gas which has expanded.

The present invention lies in the adaptation of pressure exchangers to a new use, rather than in the details of the machines themselves, of which any type may be used which lends itself to the purpose herein under discussion. Thus, for example, machines using a single rotor, or more than one rotor, may be used.

The manner in which the invention may be carried out will now be described with reference to the examples shown in the accompanying drawings in which:

Figure 1 is a longitudinal section through the engine nacelle of aircraft.

Figure 2 is a section on the line II—II of Figure 1.

Figure 2A is a part sectional elevation showing some of the ducting leaving the pressure exchanger.

Figure 3 is a similar section on the line III—III on Figure 1.

Figure 4 is a schematic view showing diagrammatically the location of the transfer gas pipes as seen from the left of Figure 1.

Figure 5 is a side elevation with the nacelle structure removed of aircraft propulsion apparatus of which the major part of the propulsive output is delivered to an airscrew.

Figure 6 is a plan view of Figure 5.

The aircraft propulsion apparatus shown in Figure 1 to 4 comprises a nacelle 1 within which is located the propulsion apparatus proper. This comprises a single rotor pressure exchanger 2 functioning as a source of high pressure hot gas for expansion, and in which the heat input is supplied by the burning of fuel in a combustion chamber 11. The pressure exchanger is scavenged at both the heat input and the heat rejection stages. The rear of the nacelle is provided with concentric nozzles 3 and 4 together forming a nozzle arrangement, while the forward end comprises an air intake duct 5 which is formed as a diffuser so that the pressure of the incoming air is raised at the expense of the velocity.

An intermediate duct 6 so constructed that it is of constant cross-sectional area connects the intake 5 with the low pressure scavenging zone of the pressure exchanger, while the low pressure gas exhausted from that zone is conducted by way of duct 9 to the nozzle 3 in which it is expanded to provide propulsive thrust which assists that of the nozzle 4 as mentioned below.

The high pressure hot gas exhausted from the pressure exchanger during the heat input stage emerges into the duct 8 which divides into two portions, of which a branch 7 conducts gas to the nozzle 4 for expansion to provide the main propulsive thrust, while a branch 10 conducts gas to the combustion chamber 11. The heat energized gas leaving the combustion chamber is returned by way of duct 12 to the input side of the high pressure scavenging zone. Fans 13 and 14 driven respectively by motors 15, 16 are provided for the scavenging gas flow, but it may

only be necessary for these fans to be used for auxiliary purposes, e. g. at starting.

Numeral **17** represents the group of transfer gas pipes which connect expansion stage cells of the pressure exchanger with cells in the compression stage. The direction of movement of the cell rotor is indicated by the arrow R in Figure 2. In order further to augment the propulsive thrust, the pressure exchanger is provided with apertures **18** (Figures 2 and 2A) which by way of ducts **18A** place cells approaching the low pressure scavenging zone **9A** into communication with the expansion nozzle **3**, so that each cell before reaching the said low pressure zone is enable to expand thus producing further propulsive thrust.

The rotor of the pressure exchanger is driven by a motor **19**. The embodiment shown in Figures 5 and 6 is similar in essentials to that of Figures 1 to 4, and common reference numerals are used for the same and similar parts. The chief difference of Figures 5 and 6 is that the gas exhausted at high pressure through the duct **7** is expanded not in a propulsion nozzle, but in a turbine **20** which through a reduction gear drives an airscrew **21** which provides most of the power needed. The turbine exhaust emerges into a duct **22** which is provided with a nozzle **23**, forming with the nozzle **3** a nozzle arrangement, so that any residual velocity in the gases may augment the thrust by expansion in the nozzle.

In both the embodiments described above, the gas flow is indicated by the arrows.

The compression of incoming air in the diffuser intake **5** has the effect of increasing the efficiency of propulsion, and also has the effect of simplifying the machine by making provision for scavenging at low pressure.

What I claim is:

1. A jet reaction propulsion unit comprising an intake for an ambient air supply, a pressure exchanger operable simultaneously to compress air from a comparatively low pressure and to expand hot gases from a comparatively high pressure, ducting by means of which air is directly fed from said intake to said pressure exchanger for compression thereby, means for providing a scavenging process for said pressure exchanger, a combustion system supplied by said pressure exchanger with compressed air for supporting combustion therein and producing high pressure combustion gases and air for expansion in said pressure exchanger, at least one nozzle located so that a fluid expanded therethrough produces jet propulsive thrust, and a direct connection from said pressure exchanger to said nozzle through which flow said expanded gases and air.

2. A jet reaction propulsion unit comprising an intake for an ambient air supply, a pressure exchanger operable simultaneously to compress air from a comparatively low pressure and to expand hot gases from a comparatively high pressure, ducting by means of which air is directly fed from said intake to said pressure exchanger for compression thereby, means for providing a scavenging process for said pressure exchanger, a combustion system supplied by said pressure exchanger with compressed air for supporting combustion therein and producing high pressure combustion gases and air, a proportion of which are expanded in said pressure exchanger, a branch passage into which the remainder of said high pressure gases and air are diverted, at least one nozzle located so that a fluid expanded therethrough produces jet propulsive thrust, a direct connection from said pressure exchanger to said nozzle through which flow said expanded gases and air, and an extension of said branch passage to said nozzle through which flow said diverted gases and air.

3. A jet reaction propulsion unit as claimed in claim 2 comprising a number of outlets from said pressure exchanger for said expanding gases and air at different stages in the expansion process and connections from each of said outlets to said nozzle arrangement.

4. A jet reaction propulsion unit as claimed in claim 2 in which said scavenging means comprises at least one fan situated in said air intake.

5. A propulsion unit delivering shaft power and jet reactive thrust comprising an air intake for an ambient air supply, a pressure exchanger operable simultaneously to compress air from a comparatively low pressure and to expand hot gases from a comparatively high pressure, ducting by means of which air is directly fed from said intake to said pressure exchanger for compression thereby, means for providing a scavenging process for said pressure exchanger, a combustion system supplied by said pressure exchanger with compressed air for supporting combustion therein and producing high pressure combustion gases and air a proportion of which are expanded in said pressure exchanger, a branch passage into which the remainder of said high pressure gases and air are diverted, a gas turbine to which said passage leads said diverted gases and air, shaft driven by said gas turbine, at least one nozzle located so that a fluid expanded therethrough produces jet propulsive thrust, a direct connection from said pressure exchanger to said one nozzle through which flow said expanded gases and air.

6. A propulsion unit as claimed in claim 5 and comprising an extension of said branch passage beyond said gas turbine to another nozzle through which flow said diverted gases and air after they have done work in said gas turbine.

7. A jet reaction propulsion unit comprising an intake for an ambient air supply, a pressure exchanger operable simultaneously to compress air from a comparatively low pressure and to expand hot gases from a comparatively high pressure, ducting by means of which air is directly fed from said intake to said pressure exchanger for compression thereby, means for providing a scavenging process for said pressure exchanger, a combustion system supplied by said pressure exchanger with compressed air for supporting combustion therein and producing high pressure combustion gases and air for expansion in said pressure exchanger, a bifurcated duct downstream of said pressure exchanger having one branch through which compressed air and gas are supplied to the combustion system and another branch through which high pressure gas is extracted from the pressure exchanger, at least one nozzle located so that fluid expanded therethrough produces jet propulsive thrust and a direct connection from said other branch to said nozzle.

8. A jet reaction propulsive unit as claimed in claim 7 in which there are two of said nozzles and another direct connection is provided between the pressure exchanger and the other nozzle for flow therethrough of said expanded gases and air.

9. A jet reaction propulsion unit as claimed in claim 8 in which said nozzles are coaxially situated with said one nozzle having its outlet upstream of said other nozzle.

10. A jet reaction propulsion unit comprising an intake for an ambient air supply, a pressure exchanger operable simultaneously to compress air from a comparatively low pressure and to expand hot gases from a comparatively high pressure, ducting by means of which air is directly fed from said intake to said pressure exchanger for compression thereby, scavenging means in said intake ducting providing a scavenging process at the heat rejection stage of said pressure exchanger, a combustion system supplied by said pressure exchanger with compressed air for supporting combustion therein and producing high pressure combustion gases and air for expansion in said pressure exchanger, further scavenging means at the heat input stage of said pressure exchanger, at least one nozzle located so that fluid expanded therethrough produces jet propulsive thrust, and a direct connection from said pressure exchanger to said nozzle through which flow said expanded gases and air.

11. A jet reaction propulsion unit comprising an intake for an ambient air supply, a rotatable pressure exchanger cell wheel operable simultaneously to compress air from

a comparatively low pressure and to expand hot gases from a comparatively high pressure, stationary ducting by means of which air is directly fed from said intake to said pressure exchanger for compression thereby, means for rotating said cell wheel, scavenging means in said intake ducting providing a scavenging process at the heat rejection stage of said pressure exchanger, a combustion system supplied by said pressure exchanger with compressed air for supporting combustion therein and producing high pressure combustion gases and air for expansion in said pressure exchanger, further scavenging means at the heat input stage of said pressure exchanger,

at least one nozzle located so that fluid expanded therethrough produces jet propulsive thrust, and a direct connection from said pressure exchanger to said nozzle through which flow said expanded gases and air.

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Sept. 11, 1956 G. JENDRASSIK 2,762,557

PRESSURE EXCHANGERS

Filed June 5, 1950 5 Sheets-Sheet 1

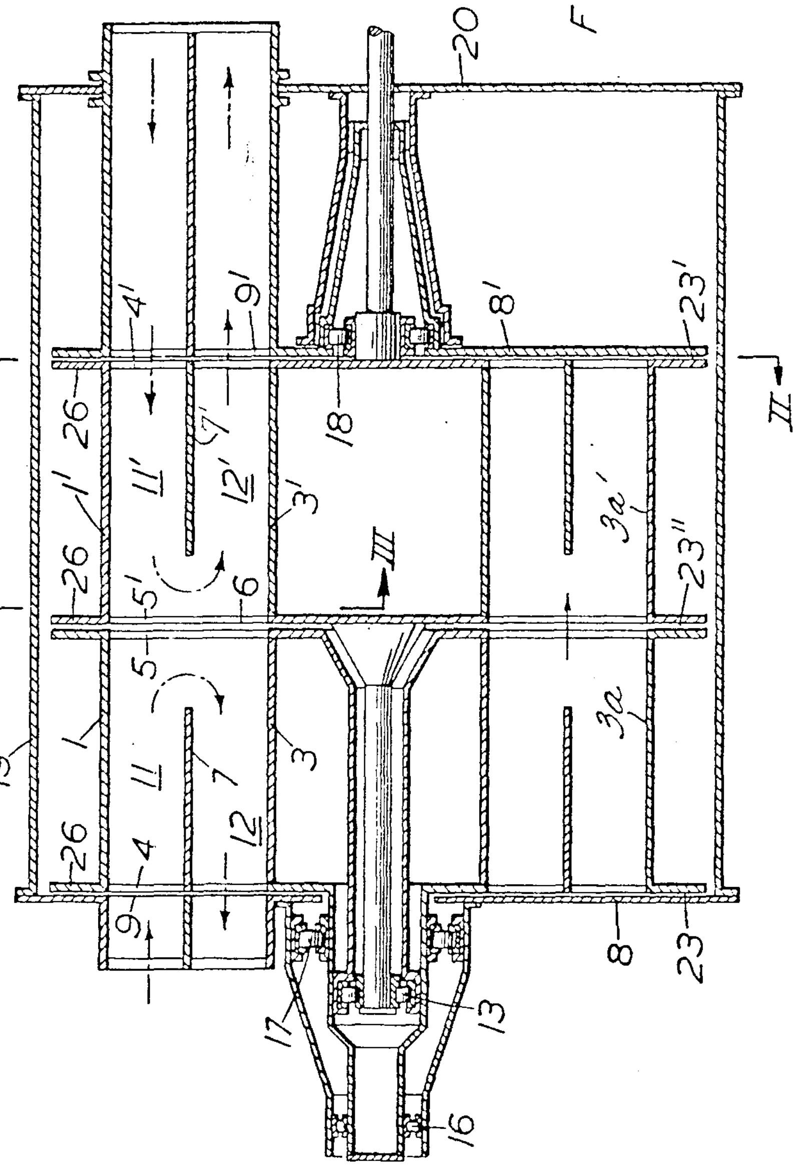

Fig. 1

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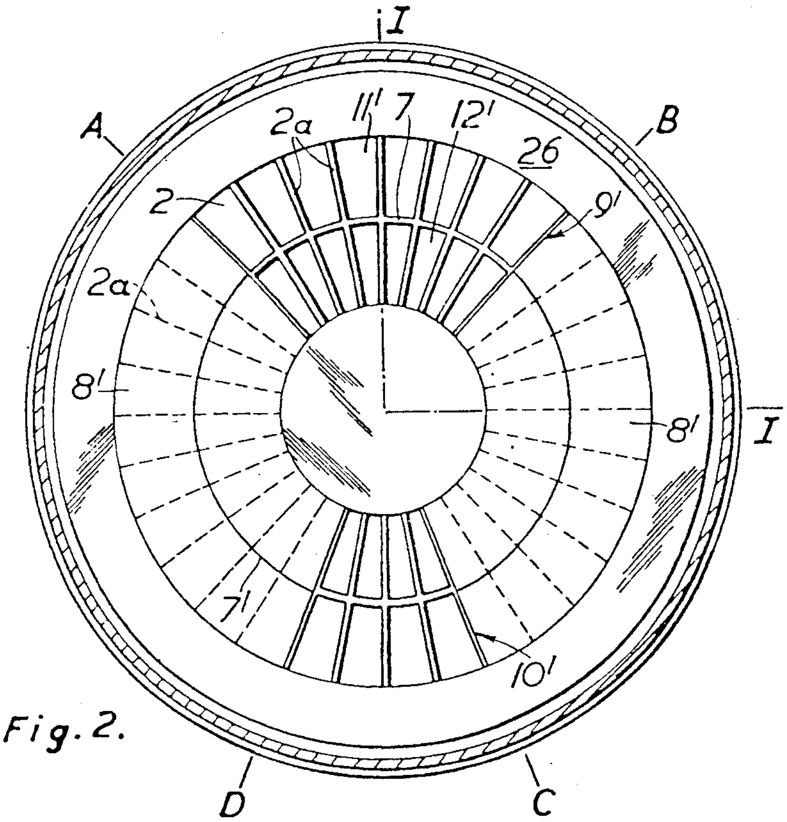

Fig. 2.

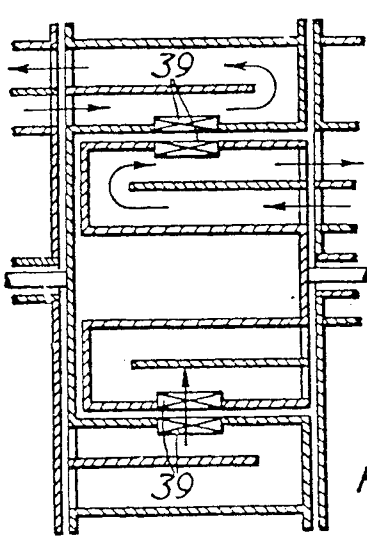

Fig. 11.

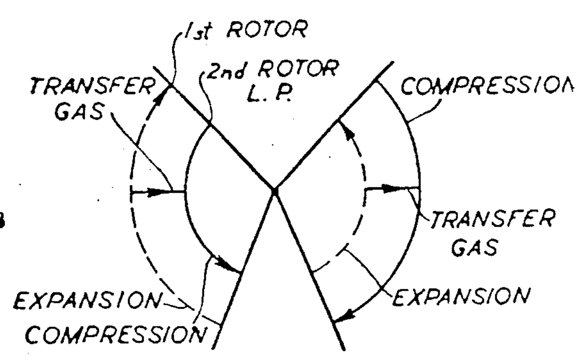

Fig. 12.

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PRESSURE EXCHANGERS

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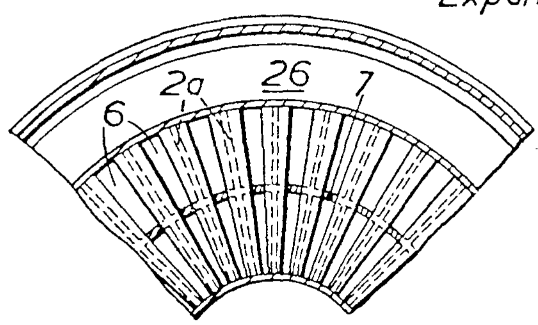

Fig. 3

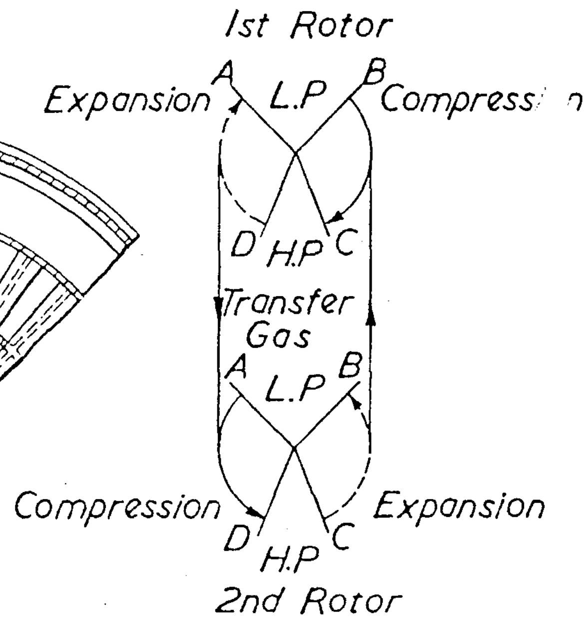

Fig. 4

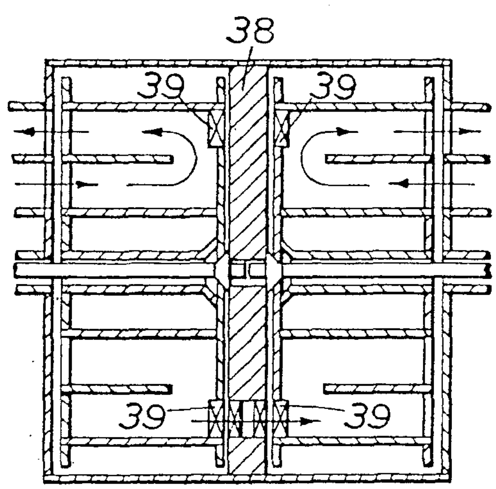

Fig. 5

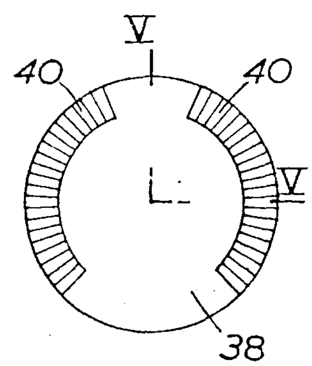

Fig. 6

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PRESSURE EXCHANGERS

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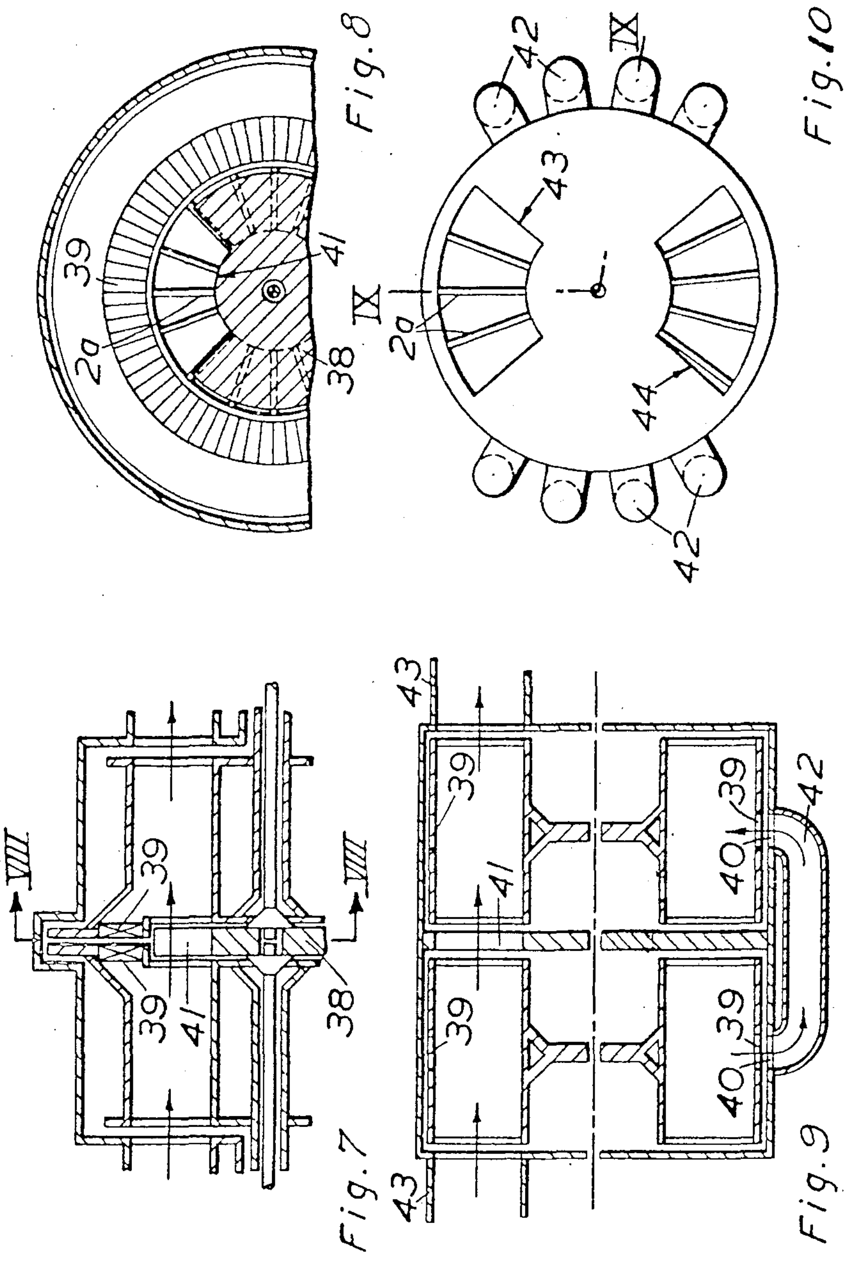

Fig. 8 Fig. 10 Fig. 7 Fig. 9

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PRESSURE EXCHANGERS

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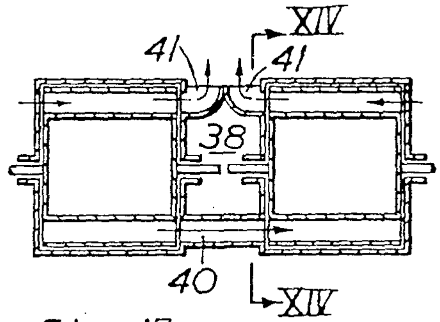

Fig. 13

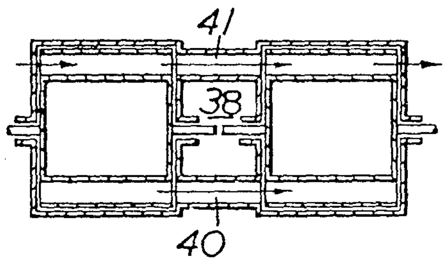

Fig. 15

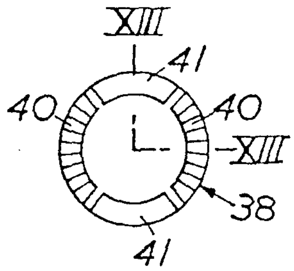

Fig. 14

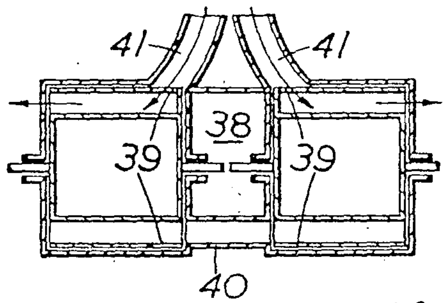

Fig. 16

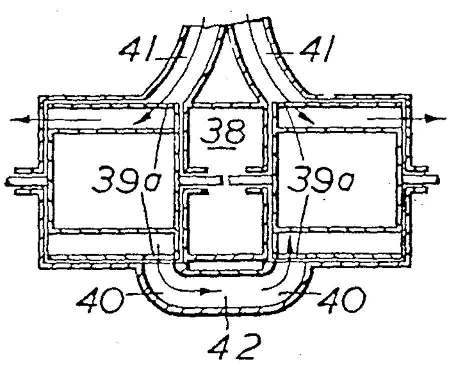

Fig. 17

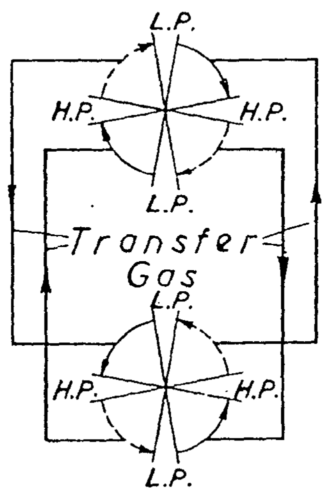

Fig. 18

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Patented Sept. 11, 1956

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PRESSURE EXCHANGERS

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Application June 5, 1950, Serial No. 166,284

Claims priority, application Great Britain June 14, 1949

8 Claims. (Cl. 230—69)

This invention relates to pressure exchangers, by which term is to be understood rotary machines (being heat engines working with gaseous fluid) which comprise at least one rotor embodying cells arranged as a circular series, the working cycle of which machines involves the compression of gas in some cells of the series and the simultaneous expansion of gas in other cells of the series, the compression and expansion stages thus formed being associated with heat input and heat rejection stages (at high or low pressure) involving the flow of gas into, and/or out of, the cells.

The gas flow arising at a heat input stage need if possible be no more than the removal from the cells of surplus gas resulting from the increase in volume which arises from heating, and similarly, the gas flow arising at a heat rejection stage need if possible be no more than the addition of gas to the cells to make up for loss in volume resulting from cooling.

In practice it is a convenient expedient for at least the heat rejection (and possibly also the heat input) to occur externally of the cells, and to this end it has been proposed for the gas flow associated with the heat rejection and heat input stages to be effected by the process (hereinafter called "scavenging") by which each cell entering the heat input (or heat rejection) zone has its gas content removed and replaced by other gas which, externally of the cells, has been specially heated (or cooled), or which in either case if derived from a source of gas which is already at the desired high (or low) temperature, this process of removal and replacement involving a continuous current of gas flowing through the cells in which it is occurring.

The points in the working cycle at which heat input and heat rejection take place depends on the intended use of the machine. If the pressure exchanger is for use as a source of high pressure hot gas (e. g. for expansion in a gas turbine, or other engine, to provide mechanical work), then heat input occurs at high pressure and heat rejection at low pressure. On the other hand if the pressure exchanger be for use as a heat pump, or a refrigerating machine, then the converse is the case (i. e. heat input at low pressure, heat rejection at high pressure).

The pressure rise (or at least a part of it) occurring in the machine may be effected by expansion stage cells being placed into communication with compression stage cells, whereby the gas in the cells at higher pressure expands into the cells at lower pressure, thus compressing gas in the latter cells, with consequent flow of some gas from expanding cells to compression cells. Such a gas flow (or one functionally corresponding to it) will be hereinafter called "transfer gas" flow to distinguish it from the gas flow associated with the heat input and heat rejection stages.

The present invention aims at providing pressure exchangers operating in a novel manner which in general are capable of offering an improved performance compared with known proposals (e. g. by rendering possible the reduction of unavoidable losses), and which may

have certain other advantages, such as constructional simplicity, higher specific output etc.

The present invention provides a pressure exchanger as above defined, comprising two rotors each embodying cells arranged as a circular series, and stator structure appropriately ported to permit the gas flow associated with heat input and heat rejection, being a machine of which the working cycle is such that, in operation, expansion cells of each rotor communicate with compression cells of the other rotor, so that, in each rotor, compression of gas is effected by the expansion of gas occurring in the other rotor, with consequent flow of transfer gas from each rotor to the other.

The invention includes arrangements having more than two rotors, of which any pair or pairs co-operate in operation as set forth in the preceding paragraph.

The rotors may be contra-rotating, and may be arranged in tandem (substantially co-axially), or otherwise. Alternatively, contra-rotating rotors, co-axially arranged, may be nested one within the other.

Scavenging (as above defined) is preferably employed at least at the heat rejection stage. Such scavenging may be in series or in parallel, with respect to the rotors.

In cases where substantially co-axial rotors in tandem are used, they may be separated by stationary structure, or they may be arranged closely adjacent with no such intervening stationary structure.

The stationary structure between the rotors, where present, may advantageously be provided with channels for the conveyance of transfer gas, and/or with port means to permit the flow of scavenging gas.

Such arrangements offer the advantage of permitting the use of larger orifices for the scavenging gas thus reducing the losses connected with scavenging, and of increasing the specific output as bigger volumes of gas can be put through the machine. The stationary channels for the transfer gas also permit the use of a better shape for the channels or the provision of guide vanes or blades in them, thus reducing the losses due to relative rotation. The stationary channels will always serve for gas flows in one direction only. Thus the shape may be adapted to the gas flow, whereas if there are no such stationary channels the available passages (since they are provided in the rotors) are required to serve for gas flow in both directions, as every cell is alternately under compression and expansion.

Further optional features according to the invention will appear below.

The invention may find useful application to pressure exchangers in which a part or all of the pressure rise and fall occurring in the machine is effected by waves of compression and expansion.

The so-called "exchange of pressure" between gas which is expanding and gas which is being compressed (e. g. between expansion cells which are in communication by "transfer gas" apertures with compression cells), which gives rise to the name of the machines under discussion, will, generally speaking, result at most in the equalisation of the pressure difference between the communicating zones. It may be possible however, by appropriate designing, to further continue the exchanging process beyond the equalisation point i. e. until the pressure of the gas which has been compressed has risen above that of the gas which has expanded.

Several examples of pressure exchangers according to the present invention will now be described with reference to the accompanying drawings. In so doing, it will be assumed for convenience that the machines described are intended for use to provide a supply of hot gas under pressure, for example, gas for expansion in a turbine to provide mechanical work. In the drawings:

Figure 1 is a section in the axial direction of one ma-

chine according to the invention, the section being taken along two different planes at right angles (as shown by the line I—I in Figure 2) so that the upper half of Figure 1 is a section through a part of the machine in which scavenging is taking place, while the lower half is a section through a part of the machine in which compression and expansion (with consequent flow of transfer gas) is taking place.

Figure 2 is a transverse section on the line II—II in Figure 1.

Figure 3 is a fragmentary view constituting a section on the line III—III in Figure 1.

Figure 4 is a cycle diagram of the operation of the machine shown in Figures 1 to 3, and other machines which are described below.

Figure 5 represents a modified machine and is an axial section taken along two planes at right angles to each other, on the lines V—V in Figure 6.

Figure 6 is a face view of the partition separating the two rotors.

Figure 7 represents a further modification and is a half section taken axially through a part of the machine where scavenging is taking place.

Figure 8 is a transverse section through Figure 7 approximately on the line VIII—VIII.

Figure 9 represents an alternative type of machine and is an axial section along planes at right angles as indicated by the line IX—IX in Figure 10.

Figure 10 is an end view of the machine shown in Figure 9.

Figure 11 shows a machine in which the rotors are arranged concentrically one within the other.

Figure 12 is a cycle diagram showing the operation of a machine according to Figure 11.

Figures 13–18 relate to further alternatives, Figures 13, 15, 16 and 17 being axial sections taken on two planes at right angles (thus Figure 13 is a section on the line XIII—XIII of Figure 14), and Figure 14 being a transverse section on the line XIV—XIV of Figure 13. Figure 18 is a cycle diagram using similar notation to Figures 4 and 12.

In Figures 1 to 3 the rotors **1**, **1′** are mounted in tandem as shown, and suitably driven to rotate in opposite directions. Each rotor **1**, **1′** comprises an inner cylinder **3**, **3′**, and outer cylinder **3*a***, **3*a*′**, and intervening radial partitions **2*a*** which form the circumferential series of cells **2**. The cells are entirely open at their outside ends, **4*a***, **4′**, except that the openings may be a little reduced in size by narrow lands (not shown) serving as mountings for labyrinth seals in order to reduce leakage from one cell to others of the same rotor, and are partly open at their adjacent ends **5**, **5′**, where the openings **6**, which permit the transfer of gas from "expansion" cells of one rotor to "compression" cells in the other rotor, only extend over about ½ the circumferential width of each cell. This arrangement prevents any cell from communicating at the same time with more than one other cell except during the scavenging periods. The cells are divided by concentric cylindrical partition walls **7**, **7′** into two compartments, the cylindrical partition extending inwards from the outer ends of the cell rotors but stopping short of the inner ends, so that the radially inner and outer compartments in every cell are in free communication in the neighbourhood of the inner ends. The outer ends of the cell rotors are covered by stationary end plates **8**, **8′** which have ports **9**, **9′** for scavenging at low pressure, and similar ports such as **10′** for scavenging at high pressure. The low pressure openings extend over an angle of about 90° (the arc A—B in Fig. 2) and the high pressure openings extend over an angle of about 45° (the arc C—D, Fig. 2). When the cells come in communication with the scavenging ports the open outer faces of the cells are fully exposed for scavenging. In the example shown, the rotors are scavenged in parallel. The ducts for the intake of fresh air to be compressed communicate with the radially outer compartments **11**, **11′** of the cells while the inside compartments **12**, **12′** communicate with the exhaust ducts for expanded gas on both outside faces of the cells, so that the scavenging air enters the outside compartment **11**, **11′**, flow axially through the cells towards the axially interior faces thereof, thus clearing the cells of exhaust gas and refilling them with fresh air, in the course of which the scavenging gas undergoes reversal of flow and returns through the radially inner compartments **12**, **12′** of the cells. This way of scavenging permits full use of the cross section of the cells. If the scavenging took place through the inner openings **6** serious throttling would occur as the average open area may not be more than ¼ of the total circumferential cross section of a cell. The high pressure scavenging takes place in a similar manner, with the difference that the flow may be reversed as compared to the low pressure scavenging, i. e. the gas may enter the radially outer cell compartments first. Compressed gas for expansion coming, it is here assumed, from a combustion chamber, may enter the radially inner cell compartments, and the gas compressed in the machine emerges from the outer compartments, whence in the present case it is assumed to be wholly or partly led to the combustion chamber, from which some gas returns to the pressure exchanger while a surplus is used to perform work, e. g. in a turbine.

In operation the cells of the one rotor leaving the high pressure scavenging ducts come into communication through apertures **6** with the cells of the other rotor travelling in the opposite direction and coming from a region of lower pressure. The gas in the cells of the first rotor, in expanding, is partly discharged through the transfer gas apertures **6**, thus effecting compression of the gas in the cells of the other rotor. Looking at Figure 2, in one rotor the cells on the arc B—C will be undergoing compression and will be individually at pressures increasing from B to C, while the cells on the arc D—A will be undergoing expansion and will be individually at pressures decreasing from D to A. In the other rotor, which is travelling in the opposite direction, the C to B cells will be simultaneously undergoing expansion while the A to D cells will be undergoing compression. The operation of the machine is diagrammatically shown in Figure 4, in which the direction of flow of the transfer gas from one rotor to the other is indicated by the arrowed straight lines. It will be noticed that there are no stationary transfer gas passages, the rotors being separated only by a small working clearance.

The cell rotors are supported as shown in bearings **13**, **16**, **17** and **18** so that the clearances **23**, **23′**, **23″** are small.

To prevent leakage losses, the end plates **8**, **8′** and the end faces of the rotors are outwardly extended in the radial direction as at **26** to provide for the accommodation of sealing means such as labyrinths or sealing blocks.

In order to keep down the leakage losses the machine may be enclosed in an outer housing **19**, **20** in which an intermediate gas pressure is maintained.

In the machine according to Figures 5 and 6, the construction as will be obvious is in general very similar to that just described above, except that the rotors are separated by a fixed central partition **38** which is provided with stationary transfer gas channels **40** (see Figure 6). Clearance corresponding to **23**, **23′**, **23″** in Figure 1 require to be small. The apertures **39** are the cell apertures for the conveyance of transfer gas and correspond in function to the apertures **6** in Figures 1 to 3.

The alternative form of machine shown in Figure 7 is one in which "straight-through" axial scavenging as opposed to "reverse flow" scavenging is used. The rotors are again separated by a stationary partition **38** which

has sector shaped ports such as 41 (Figure 8) to allow the passage of the scavenging gases. The transfer gas apertures 39 are located in a radially outward position in the adjacent ends of the two rotors as shown, so the apertures do not interfere with the scavenging flow. It will be seen that the rotors are scavenged in series.

Figures 9 and 10 show another form of machine in which straight through series scavenging is used. In this machine the rotors are again separated by a stationary partition 38 ported for scavenging flow in similar manner to the partition 38 in Figure 8. In the present case the apertures 39 for the transfer gas are provided around the outer peripheral wall of the cells, and in operation these apertures register with similar apertures 40 in the stationary casing, the apertures 40 being connected as shown by means of pipes 42 mounted externally of the casing. Alternatively it is possible to form the apertures 39 on the internal peripheral wall of the cells and to accommodate the pipes 42 internally of the casing.

Figure 11 shows a machine which functionally corresponds closely to that described with reference to Figures 1 and 2 but with the difference that the rotors are arranged concentrically one within the other, the general arrangement being obvious from the drawing. The apertures 39 for transfer gas are provided in the interior wall of the outer rotor and the exterior wall of the inner rotor. The figure is again an axial section along two planes at right angles, so that the upper half shows a scavenging zone, and the lower half shows a zone in which pressure exchange is taking place, with consequent flow of transfer gas.

Various other alternative kinds of machines using rotors in tandem are possible, using series or parallel scavenging, and in which the rotors are separated by stationary structure which may embody transfer gas channels and/or scavenging ports, or alternatively may have neither. When scavenging in parallel is employed, the direction of flow of the separate scavenging streams may be selected as convenient. Some of these further alternatives are shown in Figures 13–18, in which the reference notation is as follows:

38, stationary partition structure between the rotors.
39, special scavenging ports in outer cell periphery.
39a, special ports in outer cell periphery, used alternately for transfer flow and scavenging flow.
40, transfer gas apertures in stationary structure.
41, scavenging ports in stationary structure.
42, stationary pipes for transfer gas.

In Figures 13–16, both ends of the cells are also left open for the alternate passage of transfer flow and scavenging flow. In Figure 17, the inner ends of the cells may be closed, as shown, while the outer ends are open. The arrows in the upper half of the axial sections indicate the scavenging flow, those in the lower half indicate the transfer flow.

Some of the types of machine described above may be advantageously designed to work with each stage duplicated (i. e. for each rotor, two heat input stages, two heat rejection stages, two expansion stages and two compression stages). The working of such a machine is illustrated by Figure 18. The ducting etc. would naturally require to be duplicated accordingly.

What I claim is:

1. A pressure exchanger, in which gas compression results directly from and proceeds simultaneously with gas expansion, comprising two contra-rotatable rotors mounted on the same axis, cells around the periphery of each of said rotors, stationary structure adjacent said rotors, means providing a path through said stationary structure and cells of both rotors in series for scavenging gas flow at least once per working cycle of each cell, and means permitting transfer gas flow between cells of both rotors, when scavenging gas flow is not occurring in those cells, which transfer gas flow directly effects gas compression in cells of one rotor by gas expansion from cells of the other rotor.

2. A pressure exchanger as claimed in claim 1 in which the scavenging gas flow and the said transfer gas flow between the rotors are constrained to follow separate and distinct paths respectively.

3. A pressure exchanger as claimed in claim 2 in which paths for the said transfer gas flow are defined by apertures located in adjacent ends of the rotor and positioned radially aside from the flow path for the scavenging gases, the working clearance between the two rotors being of such smallness that, in operation, and having regard to the alignment of the said apertures, the said transfer gas flow occurs satisfactorily from one rotor to the other.

4. A pressure exchanger as claimed in claim 2 in which paths for the said transfer gas flow are defined by means comprising apertures provided in the peripheral walls of each rotor and stationary ducting so positioned that at intervals during the rotation of the rotors, the apertures are aligned with the ducting thereby forming the said paths.

5. A pressure exchanger, in which gas compression results directly from and proceeds simultaneously with gas expansion, comprising two contra-rotatable rotors mounted on the same axis, cells around the periphery of each of the said rotors, stationary structure which is adjacent the said rotors and which is ported to define a path therethrough and through the cells of both rotors in series for scavenging gas flow at least once per working cycle of each cell, and means permitting transfer gas flow between cells of both rotors, when scavenging gas flow is not occurring in those cells, which transfer gas flow directly effects gas compression in cells of one rotor by gas expansion from cells of the other rotor.

6. A pressure exchanger as claimed in claim 5 in which the rotors are mounted in tandem and are arranged to rotate at the same angular velocity.

7. A pressure exchanger as claimed in claim 6 in which the said stationary structure comprises a part separating the rotors from one another and which is ported to provide passages for the said scavenging gas flow.

8. A pressure exchanger, in which gas compression results directly from and proceeds simultaneously with gas expansion, comprising a stationary generally cylindrical structure with a part thereof extending across the interior and partitioning the said structure into two substantially equal cylindrical spaces, two contra-rotatable rotors both mounted on the longitudinal axis of the structure and each fitting closely within one of the said spaces, cells around the periphery of each rotor, ports in the end faces of the said stationary structure, corresponding passages through the said partition, together defining paths for scavening gas flow through the said structure and the rotors in series, at least once per working cycle of each cell, and channels between cells of the two rotors, by which channels transfer gas flows between cells of both rotors, when scavenging gas flow is not occurring in those cells, which transfer gas flow directly effects gas compression in cells of one rotor by gas expansion from cells of the other rotor.

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Oct. 16, 1956 G. JENDRASSIK 2,766,928

PRESSURE EXCHANGERS

Filed July 19, 1950 3 Sheets-Sheet 1

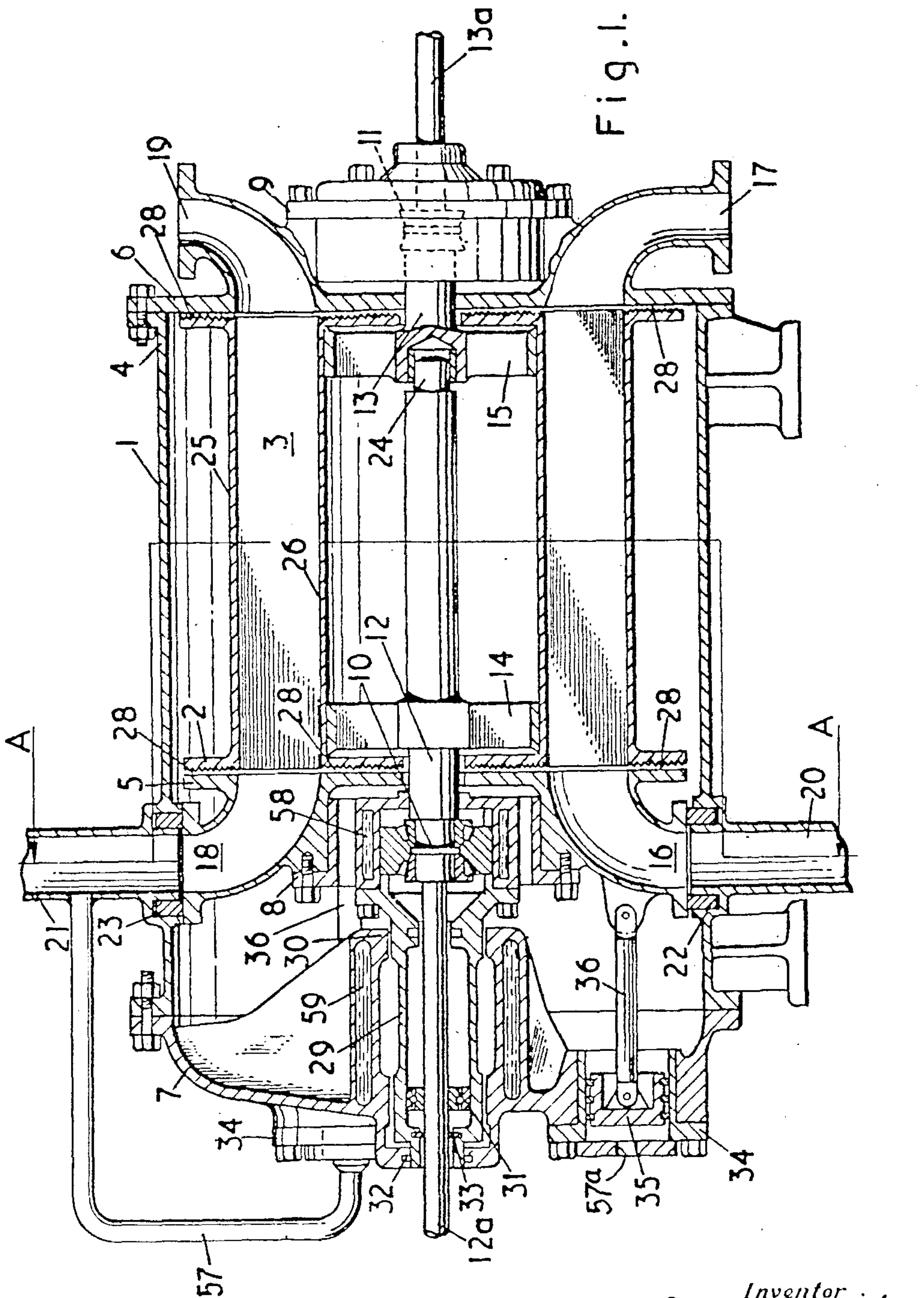

Fig. 1.

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PRESSURE EXCHANGERS

Filed July 19, 1950 3 Sheets-Sheet 2

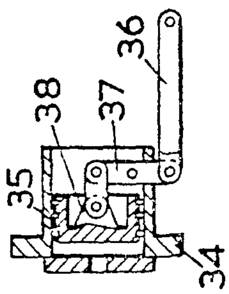

Fig. 3.

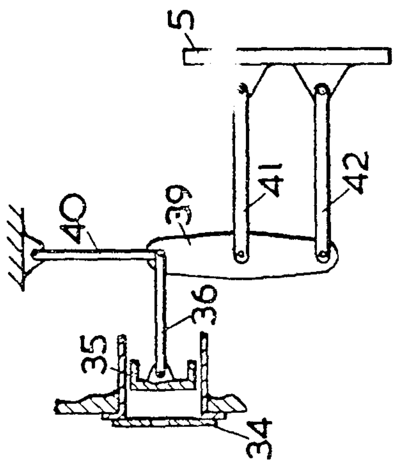

Fig. 4.

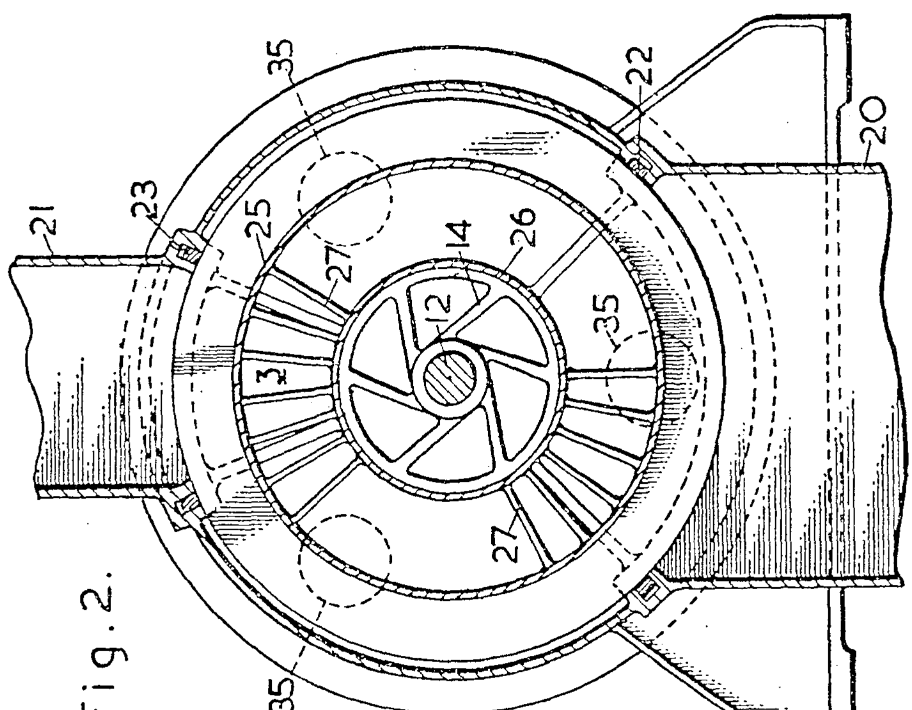

Fig. 2.

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PRESSURE EXCHANGERS

Filed July 19, 1950 3 Sheets-Sheet 3

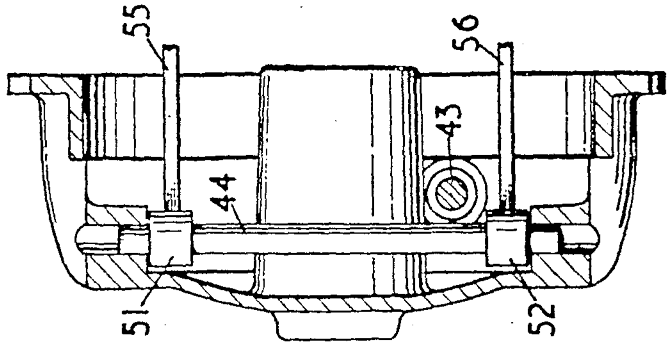

Fig. 6.

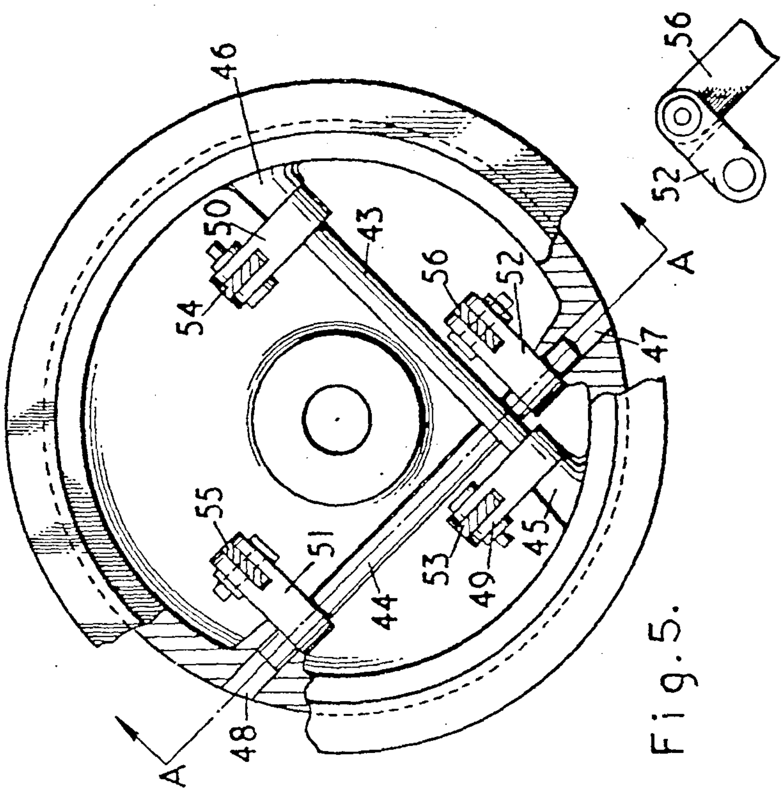

Fig. 5.

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Patented Oct. 16, 1956

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2,766,928

PRESSURE EXCHANGERS

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Application July 19, 1950, Serial No. 174,729

Claims priority, application Great Britain July 25, 1949

9 Claims. (Cl. 230—69)

This invention relates to pressure exchangers, by which term is to be understood rotary machines which compress a gaseous medium by employing the expansion, within the same machine, of another gaseous medium, or of another quantity of the same medium, the compression and expansion taking place in at least one circular series of cells between which and other structure of the machine there is a relative rotation, and the replacement of gas in the cells being effected during low pressure and high pressure scavenging stages in the former of which expanded gas in a cell is replaced by gas to be compressed, and in the latter of which compressed gas in a cell or cells is replaced by gas to be expanded.

In a typical pressure exchanger the cells are disposed to form a rotor to and from which gas is fed by non-rotary distributing means. Alternatively it has been suggested, for certain types of pressure exchanger, to use two rings of cells between which there is relative rotation, or to use a stationary ring of cells in combination with rotary gas distributing means.

Examples of known forms of pressure exchanger are disclosed for instance in British patent specification Nos. 290,669, 427,957, and 553,208, and are illustrative of the art to which the invention relates.

More particularly, the pressure exchanger applied to the present invention is of the type and operates similarly to that described in British Patent No. 553,208, corresponding to U. S. Patent No. 2,399,394. The pressure exchanger broadly includes a rotor positioned between two end plates with cells annularly arranged around its periphery and four main duct connections, through two of which gas flows toward the cells and through two of which gas flows away from the cells. In the passage of gas between the cells of the rotor and the end plates the problem of gas leakage arises. To minimise gas leakage it is essential that the clearance between such an end plate and the adjacent face of the rotor should be as small as possible, and this involves difficulties due to the thermal expansion of the rotor. The main object of the present invention is to provide constructions which permit the said clearance to be small (and to remain substantially constant) while at the same time allowing for thermal growth of the rotor in the axial direction, or for axial displacements of the rotor.

According to the present invention in a pressure exchanger for the expansion of a high pressure fluid simultaneously with compression of a lower pressure fluid comprising at least one circular series or ring of cells, at least one fluid distributing member or end plate arranged adjacent to one end face of the ring of cells for the replacement of fluid axially into the cells during relative rotation of the end plate and ring of cells, the end plate and the ring of cells are constructed and arranged adjacent one another in association with bearing means to permit relative rotation and to fix the axial spacing between them and with the end plate displaceably mounted for axial movement so as to maintain the axial spacing between

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the ring and the end plate. The bearing means may include at least one bearing for maintaining the axial spacing between the end face of the ring of cells and the end plate. The displaceable end plate may be supported by guide means permitting axial displacement but preventing tilting of the end plate relatively to the ring of cells. One form of the guide means may comprise parallel motion mechanism permitting axial displacement of the end plate without tilting and it may comprise at least two shafts pivotally supported in a plane or in parallel planes normal to the line of axial displacement of the end plate and not parallel to one another in a member relatively stationary to the end plate and each having at least two levers secured thereto which in combination with link linking them to the end plate permits displacement of the end plate without tilting. The end plate may be prevented from moving laterally by support means holding the end plate against side movement. Another form of the guide means may comprise at least one shaft or tubular sleeve secured to the displaceable end plate and extending therefrom and at least one tubular sleeve or shaft secured to a part held relatively stationary to the end plate and engaging and supporting the first mentioned shaft or tubular sleeve respectively so as to permit axial displacement of the end plate without tilting and without lateral movement.

The invention will now be described by way of example only with reference to the accompanying drawings in which:

Figure 1 is a longitudinal section through a pressure exchanger.

Figure 2 is a cross-section through the pressure exchanger on a line A—A of Figure 1.

Figure 3 is a section through a pressure balancing cylinder with a multiplying lever-linkage system.

Figure 4 is a diagrammatic view of another lever linkage system.

Figure 5 is an end view of the cover of the pressure exchanger with a parallel motion lever linkage system.

Figure 6 is a cross-sectional view of the cover with lever linkage system.

In Figures 1 and 2 a pressure exchanger **1** has a rotor **2** of drum shape containing a circular series or a ring of cells **3** rotatable within a casing **4** between two end plates **5** and **6**. The cells **3** are arranged with their longitudinal axes parallel with the axis of rotation of the ring of cells. The end plate **6** and a cover **7** are secured to the casing **4** enclosing the end plate **5** which is displaceably mounted therein so as to be axially moveable without any tendency to tilt or rotate. The two end plates **5** and **6** are provided respectively with bearing housings **8** and **9** respectively enclosing bearings **10** and **11** supporting the shafts **12**—**13** carrying the rotor **2** supported on spiders **14** and **15** adjacent to the ends of the rotor **2** and secured respectively to the shafts **12** and **13**. As the rotor is secured to the shaft **12** and the bearing, e. g. as shown the double roller bearing **10**, is fixed in position in relation to the bearing housing **8**, it follows that the axial length of the gap between the rotor **2** and the end plate **5** is substantially fixed. Expansion of the rotor results in equal axial displacement of the end plate **5** and the adjacent end face of the rotor. Similar measures apply also to the end plate **6**, shaft **13** and bearing **11**. The two end plates **5** and **6** are formed with low pressure and high pressure inlet and outlet scavenging connections **16**, **17**, and **18**, **19** through which low pressure fluid (air) and high pressure fluid (hot combustion gases) are passed for compression and expansion in the cells **3** in the rotor **2**. The connections **16** and **18** on the movable end plate **5** are arranged in sliding gas tight connection with connections **20** and **21** formed on the casing **4**. Gas tight seals **22** and **23** are

provided between the connections 16, 20 and 18, 21 respectively to prevent loss of fluid when the end plate 5 is moved. The shafts 12—13 are provided adjacent the spider 15 with an expansion joint 24 allowing for expansion and contraction of the rotor 2 and shafts 12—13 relatively to one another.

The rotor 2 is formed of two concentric cylinders, an outer cylinder 25 and an inner cylinder 26 between which radial divisions 27 extend defining the cells 3. Labyrinth seals 28 are provided between the rotor 2 and the end plates 5 and 6 to prevent leakage from the cells 3 to the spaces within the casing 4. Rubbing contact seals of high temperature heat resisting and self lubricating carbonaceous materials may be used in place of the labyrinth seals 28. The casing 4 in conjunction with the cover 7 and the end plate 6 forms a closed vessel containing the rotor 2. The spaces around the rotor 2 may be maintained at a pressure level between the high pressure and the lower pressure of the fluids being expanded and compressed into the cells 3 to diminish the pressure losses from the cells 3.

The bearing housing 8 in the movable end plate 5 is provided with an axially extending sleeve 29 extending co-axially with the rotor shafts 12—13 towards the cover 7 and located within a boss 30 extending from the cover 7 towards the bearing housing 8. The sleeve 29 carries a bearing 31 for supporting an extension 12*a* of the shafts 12—13 of smaller diameter which projects through an opening in the cover 7. The sleeve 29 is arranged to slide axially within the boss 30 and is provided with a sealing member 33 and a seal for the sleeve 29 with the shaft extensions 12*a* and a seal 32 between the sleeve and opening in the cover 7. The rotor 2 is rotated by driving the shaft through either of its ends 12*a* or 13*a* or otherwise. The extended sleeve, together with the shaft and bearings with which it co-operates, forms an anti-tilting arrangement for the end plate 5.

It is desirable that the anti-tilting arrangement should be reinforced. It will be clear from Figure 1 that, with higher pressure fluid in the upper cells than in the lower cells, the end plate 5 will experience a torque tending to turn it in an anti-clockwise direction in the plane of the figure about its bearing 10. This torque is resisted by the anti-tilting arrangement and, in addition, three pressure balancing devices are introduced. There are the three cylinders 34 (see Figures 1 and 2) formed in the cover 7 and the pistons 35 therein are connected by rods 36 to the end plate 5. The pressure has to be exerted on the pistons so that the tendency for the end plate to be axially displaced and to be tilted may be further counteracted. The pressure balancing cylinders are arranged to take some of the load off the bearing 10, to reduce the tendency for the axial displacement of the end plate and to reduce the torque on the anti-tilting arrangement which has already been mentioned. As the pressures on the end plate are not symmetrically disposed, it is clear that the pressure on the pistons will not be equal if the result desired is to be attained.

In operation the connections 21 and 19 of the pressure exchanger 1 are joined to a high pressure fluid circuit (not shown) and the connections 20 and 17 are joined to a lower pressure circuit (not shown). As air is introduced through connection 16 to at least one of the cells the right end of the cell toward which it is passing is suddenly closed by means of the rotation of the cell wheel between the end plates. A pressure wave, thus produced, starts at the closed end and proceeds with the velocity of sound toward the open end from right to left, which end is also closed when the wave reaches it. This corresponds to what happens in a well-known hydraulic ram. The pressure rise depends upon the specific weight of the gas, its speed, and the rapidity of the closing operation. When the cell is filled with compressed air it travels to the upper part of the pressure exchanger where the second phase of compression takes place. The air pressure in the cell is still below that of the pressure of the combustion gases, so that when the left-hand side of the cell is opened, to conduit 18 the right-hand side being still closed the combustion gases rush in, compressing the air with a second pressure wave to its greatest pressure. As soon as the pressure wave gets to the right-hand end of the cell, this end of the cell is opened to conduit 19 and the whole content of the cell is in movement, the air being scavenged by the gas. Now the left-hand side of the cell is closed. As the gas is still in movement an expansion wave occurs, with a corresponding reduction in pressure. When the wave reaches the right-hand end of the cell, this end is closed too and the enclosed gas, at reduced pressure, travels in the cell down toward the exit conduit 17. The last pressure drop of the gas now takes place. The cell is opened at its right-hand end and the gas expands further. As the left-hand end of the cell is still closed, another expansion wave sets in, going from right to left. The result of this wave is a kind of Kadency effect which reduces the pressure in the cell still further, thus preparing it for the reception of more fresh air. The left-hand end of the cell is now opened to conduit 16 as soon as the expansion wave reaches it. The air scavenges the gas until the cell is filled with fresh air passing from left to right. The right-hand side of the cell again closes and the new cycle begins. Thus it can be seen that end plate 5 is subjected to a high pressure load at one point and to a lower pressure load at another point. The pressure difference between the high pressure load and the lower pressure load will tend to tilt the end plate 5 whilst both loads will tend to displace the end plate 5 axially and tend to increase the clearance between it and the rotor 2. In the Figure 1 embodiment the pistons 35 are supplied with fluid pressure to balance the loads on the end plate 5. Thus high pressure fluid from an appropriate point in the pressure exchanger 1 may be applied to the two pistons 35 on either side of the high pressure connection 21 (see Figure 2) through pipe 57 whilst atmospheric pressure may be applied to the piston 35 adjacent the lower pressure connection 20 through an orifice 57*a*. In this instance the space inside the cover is assumed to be at the intermediate pressure already referred to so that piston 35 of the lower cylinder will be subject to a force to the left as shown in the figure. Hence, moments are developed by the pressure balancing cylinders which relieve the stress on the extended sleeve arrangement which is opposing tendencies for tilting or lateral movement of the end plate.

In Figure 3 there is shown a multiplying lever-linkage system 36, 37, 38 which may enable a smaller piston and cylinder to be used. In this instance, of course, the movement of the piston is reversed and the pressure applied to a piston in such an arrangement has to be appropriately adjusted in order to produce the desired resultant. In Figure 4 there is shown another way in which the pressure balancing system may be constructed. This purely diagrammatic figure indicates that only a single cylinder and piston would be required. The piston rod 36 is connected to a lever 39 which is suspended from the cover by a link 40. The end plate 5, equipped with its anti-tilting arrangement and supported on its bearing as described above, is connected to the lever 39 by appropriately positioned rods 41 and 42. As shown in the figure, and assuming a pressure tending to move the piston 35 to the right, then the rod 41 will exert a pressure to the right on the end plate whilst the force on the end plate due to the existence of rod 42 will be in the opposite direction. It will be apparent that the positioning of the rods 41, 42 may be so arranged as to provide the appropriate forces on the end plate to counteract the torque on the anti-tilting arrangement and the tendency for axial displacement of the end plate.

The movable end plate 5 may be prevented from being displaced and tilted by the arrangement shown in Figures 5 and 6 as an alternative to the extended sleeve arrange-

ment already described. The cover 7 is provided with a pair of shafts 43 and 44 which lie in parallel planes, intersect at an angle and are pivoted respectively in bosses 45, 46 and 47, 48. The shafts 43 and 44 have levers 49, 50 and 51, 52 respectively fixed to them and links 53, 54 and 55, 56 respectively connected to the end plate 5 and to the levers 49, 50 and 51, 51. Purely axial movement of the end plate causes equal longitudinal motion of the links, the respective levers and shafts turn, permitting the displacement. Tilting forces, however, try to move links connected to the same shaft in opposite longitudinal sense. Torsional forces then are induced in a shaft, which it accommodates thereby preventing any tilting. This mechanism, in taking the place of the sleeve extension 29 to the bearing housing 8 shown in Figure 1, renders the tubular boss 30 unnecessary and the shaft 12*a* emerges through a gland in the cover 7. In other respects the end plate mounting and the bearing for the end plate ar rotor are as shown in Figure 1.

Co-operating wi the anti-tilting arrangement of the pair of shafts 34 a 44 there may be arranged a single pressure balancing cylinder 34 with piston 35 and piston rod 36 to counter balance loads due to the high pressure balancing cylinder 34 and piston 35 a diaphragm motor or an expansible bellows may be used acting directly or indirectly on the end plate 5 through a lever-linkage system as described above. The pressure balancing piston or diaphragm motor or bellows is preferably disposed to act substantially in an axial direction and the pressure fluid required to operate it may be derived from an appropriate region of the pressure exchanger.

In the pressure exchanger in which it is necessary for two movable end plates to be provided between which is disposed a rotor or rotors, each end plate is constructed and arranged to be capable of axial movement to allow for thermal growth or limited axial displacement of the rotor or rotors substantially as described above for the single movable end plate. Where two movable end plates require balancing, one end plate may be balanced by one or more pressure balancing pistons, diaphragms or bellows directly whilst the other end plate may be balanced indirectly from the same pistons, diaphragms or bellows through levers and links.

Obviously where pistons movable in cylinders are used to provide the balancing force, the cylinders may be attached to the end plate and the pistons to the non-movable part of the machine which is opposite to that described and shown in the drawings.

In order to obtain good results from the invention it may be desirable to heat or cool certain parts by leading working fluid from other parts of the pressure exchanger and conducting such fluid to flow in contact with the parts such as the rotor shaft and bearings whose temperature it is desired to control, the object being to reduce the temperature gradients or to establish certain temperatures at particular parts of the exchanger.

In a pressure exchanger in which one of the fluids is at a high temperature it may be necessary to cool certain parts such as the bearings and adjacent supporting structure. In Figure 1 the bearing housing 8 is shown with a liquid cooling jacket 58 surrounding the bearing 10 and the boss 30 is shown provided with a liquid cooling jacket 59 surrounding the sleeve 29.

What I claim is:

1. A pressure exchanger comprising in combination a cell ring, an end plate and outlet means; said end plate and outlet means being mounted coaxially with said cell ring for relative rotation; said cell ring defining a series of open-ended cells extending therethrough; said end plate including inlet means on an end of said cell ring to allow the introduction of low pressure fluid into said cells, a second inlet means circumferentially displaced from said first means to allow the introduction of high pressure fluid into said cells; said outlet means being slightly circumferentially off-set respectively from said high and from said low pressure inlet means also on an end of said cell ring whereby the pressure of the low pressure fluid is raised and the pressure of the high pressure fluid is lowered in said cells during relative rotation of said cell ring, end plate and outlet means, thereby producing a pressure gradation circumferentially of the axis of rotation of said cell ring, end plate and outlet means; a supporting arrangement for maintaining at variable temperatures a predetermined clearance between the cell ring and the end plate comprising a bearing means supporting the cell ring and the end plate and a mounting supporting said bearing means and permitting said end plate to be axially displaced so that axial thermal expansion of the cell ring effects equal axial movement of the end plate.

2. In a pressure exchanger a supporting arrangement as claimed in claim 1 in which said bearing means comprises a thrust bearing, bearing housing and shafting carried by said thrust bearing which support both cell ring and end plate.

3. In a pressure exchanger a supporting arrangement as claimed in claim 2 in which said mounting comprises anti-tilting means for counteracting the tendency of the end plate to tilt due to unequal pressure being applied to diametrically opposed portions of the end plate.

4. In a pressure exchanger a supporting arrangement as claimed in claim 3 in which said anti-tilting means comprises a parallel motion mechanism.

5. In a pressure exchanger, a supporting arrangement as claimed in claim 4 in which said anti-tilting means comprises a member interconnected with said end plate and stationary relative thereto, at least two shafts in parallel planes normal to the line of axial movement of the end plate and not parallel to one another pivotally supported in said stationary member, two levers secured to each shaft and rigid links connecting each lever to the end plate.

6. A pressure exchanger comprising in combination a cell ring, an end plate and outlet means; said end plate and outlet means being mounted coaxially with said cell ring for relative rotation; said cell ring defining a series of open-ended cells extending therethrough; said end plate including inlet means on an end of said cell ring to allow the introduction of low pressure fluid into said cells, a second inlet means circumferentially displaced from said first means to allow the introduction of high pressure fluid into said cells; said outlet means being slightly circumferentially off-set respectively from said high and from said low pressure inlet means also on an end of said cell ring whereby the pressure of the low pressure fluid is raised and the pressure of the high pressure fluid is lowered in said cells during relative rotation of said cell ring, end plate and outlet means, thereby producing a pressure gradation circumferentially of the axis of rotation of said cell ring, end plate and outlet means; a supporting arrangement for maintaining at variable temperatures a predetermined clearance between the cell ring and the end plate comprising a bearing means supporting the cell ring and the end plate, a member interconnected with said end plate and stationary relative thereto, a co-axially extending shaft and sleeve combination one part of which is secured to and extends from the end plate while the other part is secured to and extends from said stationary member, this combination opposing any tilting tendency of the end plate, and a mounting supporting said bearing means and permitting said end plate to be axially displaced so that axial thermal expansion of the cell ring effects equal axial movement of the end plate.

7. A pressure exchanger comprising in combination a cell ring, an end plate and outlet means; said end plate and outlet means being mounted coaxially with said cell ring for relative rotation; said cell ring defining a series of open-ended cells extending therethrough; said end plate including inlet means on an end of said cell ring

to allow the introduction of low pressure fluid into said cells, a second inlet means circumferentially displaced from said first means to allow the introduction of high pressure fluid into said cells; said outlet means being slightly circumferentially off-set respectively from said high and from said low pressure inlet means also on an end of said cell ring whereby the pressure of the low pressure fluid is raised and the pressure of the high pressure fluid is lowered in said cells during relative rotation of said cell ring, end plate and outlet means, thereby producing a pressure gradation circumferentially of the axis of rotation of said cell ring, end plate and outlet means; a supporting arrangement for maintaining at variable temperatures a predetermined clearance between the cell ring and the end plate comprising a bearing means supporting the cell ring and the end plate, at least one fluid operated pressure balancing device acting upon the end plate to reduce the load upon said bearing means and a mounting supporting said bearing means and permitting said end plate to be axially displaced so that axial thermal expansion of the cell ring effects equal axial movement of the end plate.

8. In a pressure exchanger, a supporting arrangement as claimed in claim 7 in which said pressure balancing device comprises a cylinder, a piston head sliding in the cylinder and a connection through which fluid has access to the piston head.

9. A pressure exchanger comprising in combination a cell ring, an end plate and outlet means; said end plate and outlet means being mounted coaxially with said cell ring for relative rotation; said cell ring defining a series of open-ended cells extending therethrough; said end plate including inlet means on an end of said cell ring to allow the introduction of low pressure fluid into said cells, a second inlet means circumferentially displaced from said first means to allow the introduction of high pressure fluid into said cells; said outlet means being slightly circumferentially off-set respectively from said high and from said low pressure inlet means also on an end of said cell ring whereby the pressure of the low pressure fluid is raised and the pressure of the high pressure fluid is lowered in said cells during relative rotation of said cell ring, end plate and outlet means, thereby producing a pressure gradation circumferentially of the axis of rotation of said cell ring, end plate and outlet means; a supporting arrangement for maintaining at variable temperatures a predetermined clearance between the cell ring and the end plate comprising a bearing supporting said end plate, shafting supporting said cell ring and running in said bearing, a telescopic joint in said shafting to allow for axial expansion of the cell ring and a mounting supporting said bearing and permitting said end plate to be axially displaced so that axial thermal expansion of the cell ring effects equal axial movement of the end plate.

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PRESSURE EXCHANGERS AND APPLICATIONS THEREOF

Filed Nov. 5, 1952 3 Sheets-Sheet 1

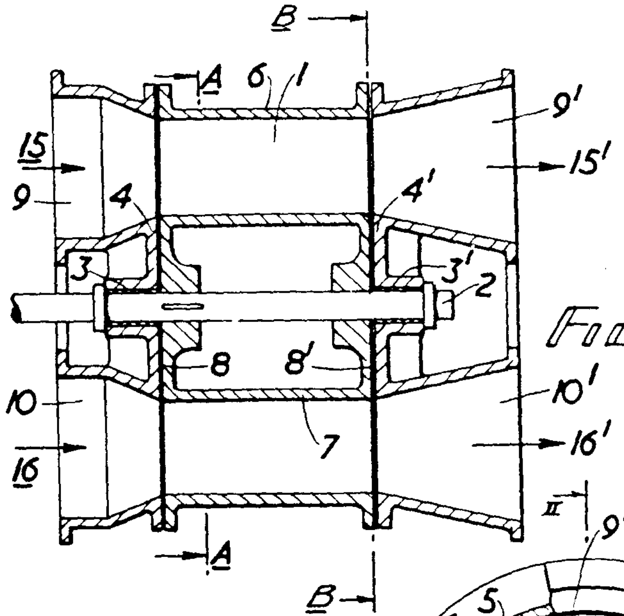

FIG. 1.

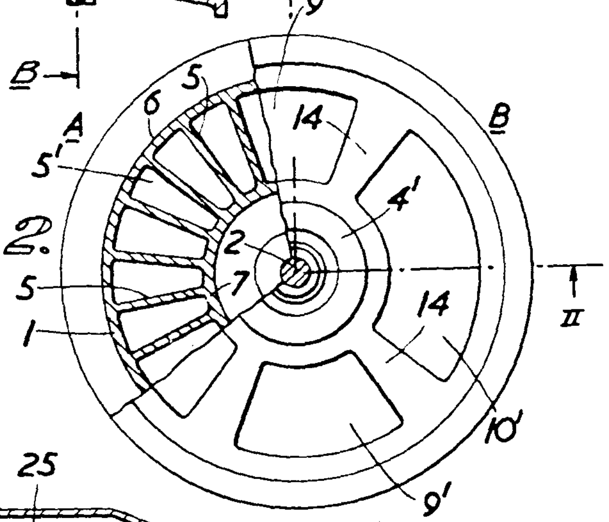

FIG. 2.

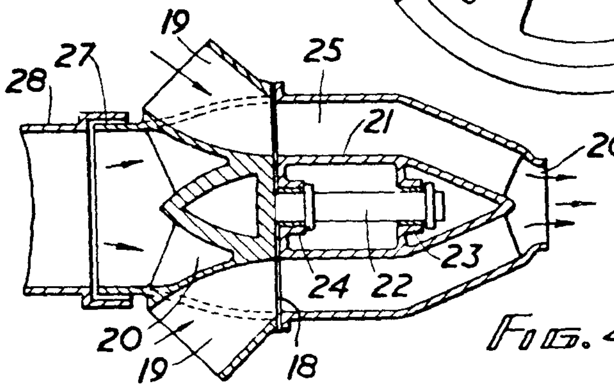

FIG. 4.

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PRESSURE EXCHANGERS AND APPLICATIONS THEREOF

Filed Nov. 5, 1952 3 Sheets-Sheet 2

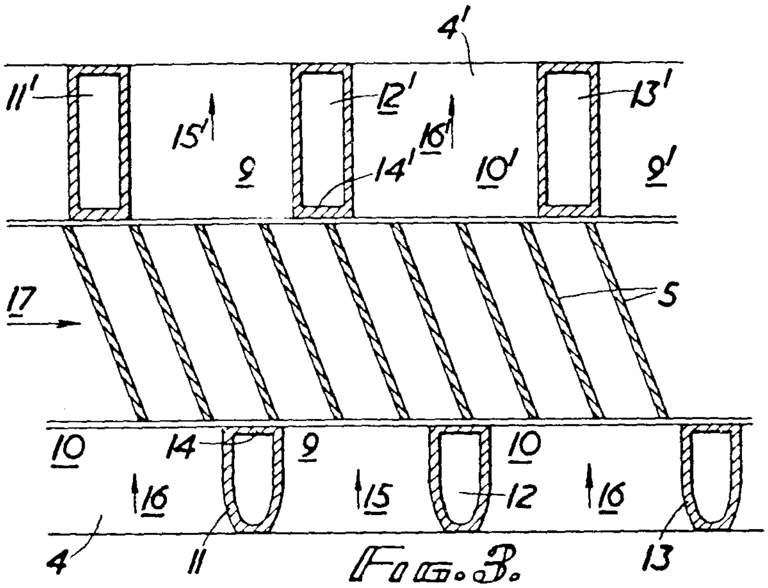

FIG. 3.

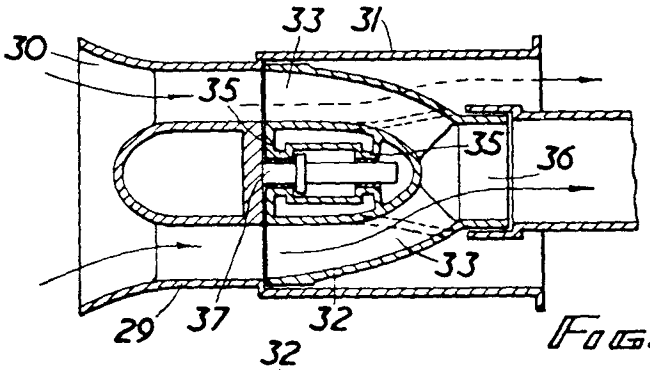

FIG. 5.

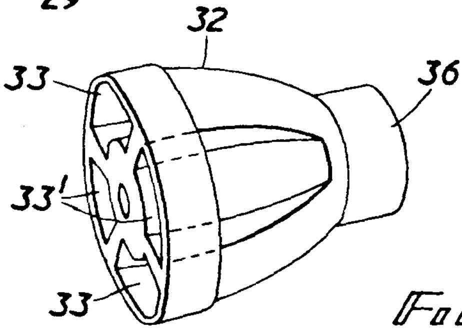

FIG. 5A.

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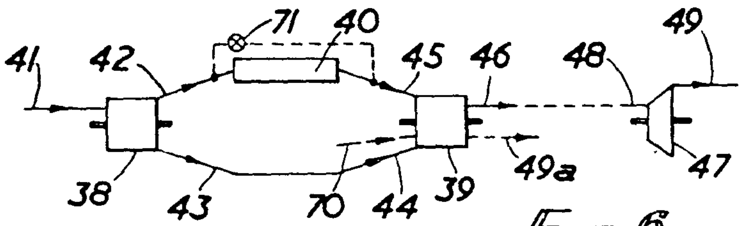

FIG. 6.

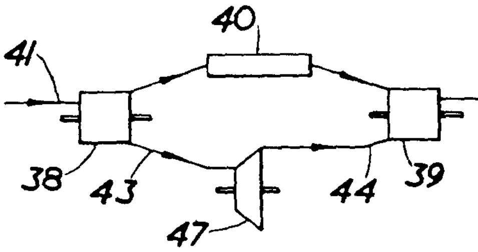

FIG. 7.

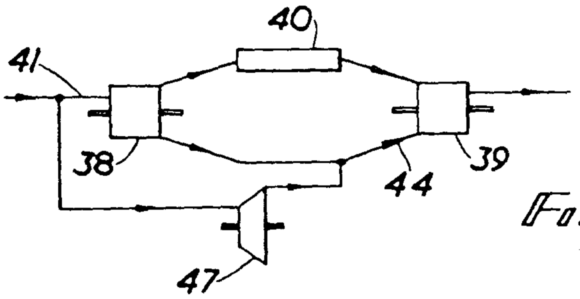

FIG. 8.

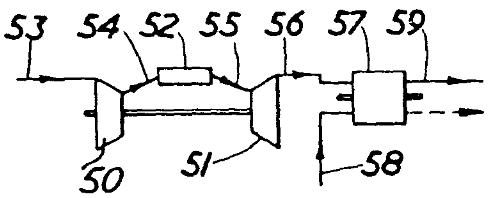

FIG. 9.

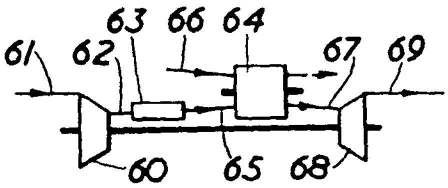

FIG. 10.

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2,946,184
Patented July 26, 1960

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PRESSURE EXCHANGERS AND APPLICATIONS THEREOF

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Filed Nov. 5, 1952, Ser. No. 318,919

Claims priority, application Great Britain Nov. 8, 1951

12 Claims. (Cl. 60—35.6)

This invention relates to pressure exchangers and to applications thereof. Pressure exchangers have been previously proposed in which work is transmitted from a fluid of higher pressure level to a fluid at a lower pressure level. Suggested apparatus for this purpose has comprised a cell wheel with end plates adjacent the ends thereof through which fluid is delivered to and discharged from the cells. In such apparatus the cell wheel and end plates are relatively rotatable. In one proposed pressure exchanger the transmission of work capacity was to be performed by fluid pressure impulses within the cells whilst in others the expansion of fluid from one cell effected compression of fluid in another. Scavenging through the cells at one stage of operation is a common expedient.

The present invention provides a pressure exchanger comprising a number of cells each of which is placed in communication at its ends with fluids at different pressure levels so that there is at such times a pressure difference between the two ends of the cell and so that, by acceleration and deceleration of fluid within the cells, work is transmitted between fluid communicating with the pressure exchanger at the different pressure levels.

The cells of such a pressure exchanger may conveniently be arranged annularly around the periphery of a cell wheel which is rotatable relatively to an adjacent end plate. Through that end plate the cells may have access to a plurality of ducts circumferentially spaced relative to the axis of rotation, through which fluids at different pressure levels are delivered to the cells. At the other end of the cells from that end plate communication may be obtained to one or more ducts maintained at pressure levels different from those of the delivered fluid; a similarly arranged end plate may be incorporated at this other end also. The width of the duct passages through an end plate and their mutual relationship are preferably arranged so that in a normal operation most advantageous use is made of acceleration and deceleration impulses caused in the fluid within a cell as the latter moves relatively to the duct passages.

A pressure exchanger according to the invention is stated above to have in operation a pressure difference between the two ends of a cell. In certain instances it is convenient to transform a fluid pressure into a velocity component, for example one end of a cell may be deliberately made convergent in cross sectional area so that fluid is discharged therethrough at a higher speed than it otherwise would be. Such an increase in velocity at the expense of fluid pressure and similar transformations are intended to be maintained within the scope of the invention.

The present invention further provides a pressure exchanger comprising a number of cells each of which in turn is placed in communication at one end with fluid at two different pressure levels alternatively and at its opposite end only with fluid at a pressure level between said different pressure levels.

As an extension of the latter pressure exchanger each

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of the cells at its said one end may be placed in communication with a plurality of fluid sources at different pressure levels in succession and at its opposite end only with fluid at a pressure level intermediate between the extremes of said pressure levels.

According to the invention there is provided a pressure exchanger which is arranged substantially to equalize the pressure of fluids at a plurality of pressure levels comprising a cell wheel, and end plate adjacent thereto, means for effecting relative rotation between the cell wheel and the end plate, a plurality of ducts through each of which one of said different pressure fluids is conveyed to the end plate for delivery to the cell wheel, the arrangement being such that every cell in turn is placed in communication at one end with said ducts in succession, and a common duct whose internal pressure is maintained at a level intermediate between the extreme pressures of said delivered fluids into which fluid is discharged from the other end of the cells.

The latter is a special form of pressure exchanger, that is one which receives fluids at different pressure levels and in effect discharges fluid at an intermediate pressure level, this apparatus may be termed a pressure equalizer.

The invention also provides a pressure exchanger which is arranged to produce from a fluid delivered thereto supplies a fluid at a plurality of pressure levels, at least one being higher and another lower than the pressure of said delivered fluid, comprising a cell wheel, an end plate adjacent thereto, means for effecting relative rotation between cell wheel and end plate, a duct through which said delivered fluid is passed to one end of every cell and a plurality of ducts extending downstream of the cell wheel from the end plate each of which ducts is maintained at a different internal pressure level at least one being higher and another lower than the pressure of said delivered fluid and into which ducts fluid is discharged from the other end of the cells.

The latter is another special form of pressure exchanger, that is one in which delivered fluid transmits work to fluid leaving at a higher pressure, further fluid leaving at a lower pressure. This form of pressure exchanger may be termed a pressure divider.

Pressure exchangers according to the present invention have several distinguishing features from those according to the prior art. For instance embodiments of the present invention function as a result of practically continuous presence of acceleration or decleration fluid pressure impulses within any cell. There is no appreciable cell scavenging taking place at any stage in the operation of the apparatus. Moreover the prior proposals have been always for pressure exchangers forming the major part of either a heat engine or a heat pump. It has been a common factor of previously suggested pressure exchangers for fluid extracted from the cell wheel of the exchanger to be reintroduced, e.g. after heating, into the cell wheel. This feature is not present in embodiments of the present invention.

In this case, as with the prior art, the fluids between which work is transmitted are most conveniently in gaseous state. However pressure exchangers according to the present invention are not exclusively intended to operate upon gases, liquids may also be employed and it is possible to envisage embodiments intended for work transmission between liquid and gaseous fluids.

Pressure exchangers as now set forth have many applications, particularly as components of heat engine plant.

The present invention provides a heat engine comprising a first pressure exchanger receiving gas at a first pressure level and discharging said gas at two pressure levels one higher and the other lower than said first level, a gas heating system receiving gas at said higher pressure level from the first pressure exchanger and discharging

hot gas and a second pressure exchanger receiving said hot gas and gas discharged at said lower pressure level from the first pressure exchanger and discharging a stream of hot gas.

The hot gas stream, the gas at the lower pressure level of the first pressure exchanger or further gas received at the first pressure level may be expanded through an expansion machine. The latter is preferably a gas turbine.

If the second pressure exchanger is arranged to receive a further gas supply at a low temperature compared with the hot gas, greater heat may be introduced via the heating system. It is possible by replacement of the heating system with an internal combustion engine to supercharge the latter.

The expanded hot gases leaving a conventional compressor-gas turbine set may conveniently be introduced into a pressure exchanger with a lower pressure gas supply. The discharged gas therefrom may then be expanded through a nozzle to produce a highly efficient propulsive thrust.

In another possible arrangement of a compressor-gas turbine set, hot gas leaving the heating system to the set is introduced into a pressure exchanger together with a lower pressure gas supply. The gas discharged from the pressure exchanger is then expanded through the turbine. By this arrangement the heating system can be allowed to introduce more heat than otherwise would be the case.

The invention will now be described with reference to certain embodiments thereof shown by way of example only in the accompanying drawings in which:

Figure 1 is a cross-sectional elevation through a pressure exchanger according to one embodiment of the invention, the upper half being a section at right angles to the lower half and shown as section II—II of Figure 2.

Figure 2 shows in its sector A a cross-section on the line AA of Figure 1 and in its sector B the face of an end plate of the same embodiment viewed at the position BB. i.e. from the right hand end of the cell wheel structure **1** of Figure 1.

Figure 3 is a peripheral development of part of the cell wheel and ducting making up the embodiment of Figure 1.

Figure 4 is a cross-sectional elevation through a second embodiment of the invention and Figure 5 shows a corresponding view of a third embodiment of the invention.

Figure 5A is a perspective view of a component of the apparatus of Figure 5.

Figures 6, 7 and 8 show three alternative ways in which pressure exchangers according to the invention can be incorporated into new heat engine arrangements in order to achieve certain desired advantages.

Figure 9 shows the incorporation of a pressure exchanger according to the present invention in a gas turbine jet propulsion engine.

Figure 10 shows an alternative arrangement of gas turbine plant lay-out in which the position of the pressure exchanger is different.

Turning first to the embodiment shown in Figures 1 to 3 there will be seen a rotary cell wheel **1** carried by a shaft **2** which is itself located in bearings **3** and **3'**. The bearings are nested within end plates at opposite ends of the cell wheel as shown at **4** and **4'**. The rotary cell wheel consists of a cylinder with a peripheral annulus formed between the outer wall **6** and the inner wall **7**, which annulus is divided by axial/radial partitions **5** (which can best be seen in Figure 2) thus forming a number of cells **5'**. The end faces **8** and **8'** of the cell wheel are located closely adjacent to the respective faces of the end plates **4**, **4'** and sealing arrangements (not shown) are provided between the cell wheel faces and the end plates to reduce leakage losses. Such seals may take the form of labyrinth glands or sliding seals for example. They need not necessarily be on opposing faces in the radial plane as shown in Figure 1 but can quite well be provided on cylindrical surfaces, that is co-axial with the rotor and in some circumstances this may be preferable. In Figure 3 opportunity has been taken to show that the partitions **5** making up the cells **5'** of the cell wheel need not necessarily be axial but can be inclined to the axial plane. Each end plate has ducts extending through it away from the cell wheel through which fluid is arranged to pass to and from the cell wheel. The duct passages **9**, **9'**, **10**, **10'**, in the end-plates are arranged in circumferentially spaced fashion around the shaft axis and are separated from one another by partitions **11**, **11'**, **12**, **12'**, **13**, **13'**, shown clearly in Figure 3.

In the faces of the end-plates adjacent to the cell wheel there are sector shaped wall sections such as that shown at **14** in Figure 2 which divide the duct passages from one another. The passages themselves, such as that shown at **9**, may be formed either as convergent or divergent passages as is required by the design of the pressure exchanger. In Figure 1 the arrows **15**, **15'**, **16**, **16'**, are intended to show the direction of flow of gas through the pressure exchanger and in Figure 3 will be seen an arrow **17** which indicates the direction of rotation of the cell wheel relative to the stationary end plates. In this embodiment it is the cell wheel that rotates but it is clearly apparent that relative rotation may be obtained between cell wheel and end plates by an arrangement of end plates which itself rotates, the cell wheel being a stationary structure. As shown in Figure 1 the rotating cell wheel mounted upon the shaft **2** may be driven by an external drive connected to the free end of the shaft.

The end plates **4** and **4'** can be secured to each other as a rigid structure but they are preferably arranged relative to the cell wheel so that there is allowance for expansion. The supporting arrangement then has to be such as permit axial displacement of the end plates relative to one another without any tilting being permitted. This is required so that the sealing system between end plates and cell wheel may be maintained. An arrangement has been previously described in U.S. Patent Application Serial No. 323,490, now U.S. Patent 2,779,530 in which the clearance gap between cell wheel and end plate is kept substantially constant in spite of thermal expansion. Use was made of bearings which are capable of taking axial thrust and are arranged closely to the clearance gap which is to be maintained. Tilting of the end plate was also prevented by the provision of guiding means which allows the end plate to be displaced in an axial direction only.

The operation of the pressure exchanger shown in the first three figures is as follows:

It will be recalled that in Figure 1 the ducts **9** and **10**, and **9'** and **10'** are alternately arranged around the shaft axis, in fact approximately at right angles to one another as is more clearly shown in the sector B of Figure 2. The two diametrically opposite ducts **9** are bifurcated upstream of the pressure exchanger from a single duct. The same arrangement applies to the inlet passages **10**. These are also bifurcated branches of a single duct. The duct leading to the inlet passages **9** communicates with the source of fluid at one pressure and the duct leading to the inlet passages **10** communicates with a source of fluid at another pressure. One of these pressures may conveniently be atmospheric pressure the fluid supplied being ambient air. The outlet passages from the pressure exchanger **9'** and **10'** are either each connected to sources of fluid at different pressures or are all connected to one single outlet duct downstream of the pressure exchanger. In the latter case the outlet duct is itself maintained at a pressure intermediate between the pressure levels of the two inlet fluid supplies.

It will be seen that the low pressure fluid entering the cell wheel does so by means of passages which alternate with passages for the entry of fluid at a higher pressure.

Different pressure levels exist at the other end of the cells, e.g. two pressures between the higher and lower fluid entry pressures. Thus the cells receive impulses of high pressure fluid causing acceleration in each cell followed by periods of depression where the low pressure fluid acts on a cell thereby effecting deceleration therein. At the other end of each cell fluid contained therein is delivered to the pressure level obtaining in the adjacent duct passage, the entering fluid of higher pressure in effect pumping or compressing the entering fluid of lower pressure which also enters, to the two leaving pressure levels. Thereafter the latter fluids are used to provide reactive thrust or expanded in an expansion machine or employed in some other convenient way.

The relationship between the spacing of the cell partition walls 5 and the width of the ducts 9, 9′, 10, 10′, has been previously discussed in relation to other pressure exchanger proposals. As shown in Figure 3, e.g. at 14 and 14′, the end plate walls have distinct parts which provide duct passages from one another and which provide that communication between any cell and the ducts shall be intermittent. It is preferred that this should be so even when one end of all the cells is opened only to a common fluid pressure level. The duct passage openings in the end plate walls are preferably so located that an end of a cell is closed by the end plate wall substantially at the instance of arrival at that cell end of a compression or expansion impulse caused by the other end of the cell being closed. It is also preferable that an impulse caused by an end of a cell being opened to a duct passage should reach the other end of the cell at the instant when that other cell end is opened to its passage. Although it is not possible to define as closely as one would wish the instant at which a cell passes a duct passage edge and thereby gives rise to an impulse in the cell, it is desirable that a cell should remain in contact with a duct for a period equal to the time required for an impulse to travel from one end of a cell to the other and back again or an integral multiple thereof.

The parts of the end plate walls between various duct passages leading to and from the cell wheel may not be of equal width due to the fact that depression impulses become extended in length when travelling through the cell and compression impulses are decreased in length. Considering this in connection with Figure 3 there will be seen there an end plate structure 11 having a wall adjacent the cell wheel 14. At the other end of the cell wheel there is the end plate having a part of the wall between adjacent passages shown at 14′. The impulse due to the closing of a cell by the end wall part 14 has to reach the other end of the cell at the moment when the end plate wall 14′ is closing the same cell. However, when a cell passes the end plate wall 14 and is opened thereby the impulse travelling to the other end of the cell has to reach there at the same moment when the cell is being opened by the end plate wall edge 14′. As the rotation of the cell wheel is at constant angular velocity and the times for the impulses to travel through the cell are different the widths of the end plate wall sectors 14 and 14′ are preferably themselves different for efficient operation.

Consider higher pressure fluid entering into the pressure exchanger of Figure 3 in the direction of the arrows 16 through the entry passage 10. In the left hand cell of the peripheral development of the cell wheel there is fluid at a pressure lower than the fluid entering through the passage 10, and its other end opening to passage 9′ is also at a lower pressure hence the fluid in the cell is accelerated. The higher pressure fluid entering the cell sets up an acceleration impulse which is able to pump the fluid in the cell to the pressure level obtaining in the duct 9′ which is assumed to be higher than that in duct passage 10′. Fluid therefore leaves the cell in the direction of the arrow 15′. On connection of the same cell to the outlet passage 10′ and to the inlet duct 9 where a lower pressure obtains the pressure difference between the ends of the cell changes sign and retardation of the fluid flow is caused. However a depression exists at the upstream end of a cell as it becomes opened to passage 9 and fluid continues to enter the pressure exchanger. At the downstream end of the cell fluid is discharged into the passage 10′ in the direction 16′. The cell proceeds on its way and once again it becomes open at its upstream end to a fluid of higher pressure through the next duct 10 and an acceleration impulse pumps the fluid in the cell out through the next outlet passage 9′.

The high pressure fluid supplied to the pressure exchanger may conveniently be the hot combustion products from a combustion system. Where this is so and the lower pressure fluid is atmospheric air there is a very useful application of this machine to gas turbine plant in a manner which will be described in more detail below.

In the embodiment of the invention illustrated in Figure 4 it will be seen that there is only one end plate proper, namely that shown at 18 which carries ducts 19 and 20 for fluids at two different pressure levels entering the pressure exchanger. In this diagram it is more clearly shown how the entry duct 20 is bifurcated upstream of the pressure exchanger, fluid supplied therefrom going to diametrically opposed duct passage openings in the end plate adjacent to the cell wheel. The cell wheel rotor 21 is supported by a shaft 22 which runs in bearings 23 and 24. The cells 25 themselves are of more complicated from than those shown in the embodiment above in that they are partly conical. Their downstream ends open into a common discharge orifice 26 through which fluid leaves the pressure exchanger. As in the previously described embodiment two fluids at higher and lower pressure levels enter the device by the ducts 19 and 20 respectively which alternate peripherally around the annular entry into the cells 25 through the end plate 18. The common duct terminating in the discharge orifice 26 is in this case maintained at a pressure intermediate between the two inlet pressures and the fluid leaves the cells at that pressure. Pressure is immediately transformed into velocity but in operation it can be considered that a pressure difference exists across the ends of each cell. An embodiment very similar to that shown in Figure 4 can be clearly envisaged in which it is the end plate structure which rotates to a stationary cell wheel. Such a possibility is illustrated by the mating together of the rotating and stationary parts of the inlet duct 20 by the peripheral surface 27 running within the stationary bearing surface at the end of the duct 28. This arrangement will not only permit rotation but will take up axial displacement of the end plate. A labyrinth seal or the like would be included in this arrangement. If it is so desired the orifice 26 may be shaped as a nozzle so that the fluid leaving the pressure exchanger can expand and accelerate.

While the two embodiments described above have been pressure exchanging and pressure equalizing arrangements, that shown in Figure 5 is a pressure dividing arrangement. A rotary cell wheel 29 is formed integrally with the ducting through which fluid is passed to the cell wheel. Fluid enters the cell wheel in this case at one pressure level only. Also integral with the cell wheel itself is a cylindrical duct 31 into which is discharged fluid at one of the two outlet pressure levels. In this instance it is the end plate which is rotatable relative to a stationary cell wheel, the end plate being shown at 32. It comprises ducts 33 for fluid leaving the pressure exchanger at a different pressure from that of fluid leaving through the cylindrical duct 31. The end plate is supported by a shaft 37 which runs in bearings 35. The outlet duct passages 33 join into a common conduit 36 in which there is maintained a pressure level either higher or lower than the input pressure level.

In Figure 5A there will be seen a diagrammatic perspective view of the end plate 32, showing in particular the arrangement of the outlet ducts. The two ducts 33 in the upstream end of the end plate allow fluid to pass through to the common outlet conduit 36. The two other diametrically opposed ducts 33' allow fluid to pass to the outside of the end plate 32 halfway along its length. Referring back to Figure 5 it will be seen that this allows the fluid taken through these ducts to be discharged through the cylindrical duct 31. The duct 31 and the conduit 36 are maintained at two different pressure levels one above and the other below the input pressure level to the pressure exchanger.

The embodiment shown in Figure 5 therefore provides a pressure divider one part of the fluid entering into the pressure divider having imparted work to another part of the same fluid. Fluid enters the pressure divider at a medium pressure and is delivered at the other end of the cell wheel into alternatively arranged duct passages one group being at a higher pressure and the other at a lower pressure than the entering pressure. When the downstream end of any cell is connected to the lower pressure outlet duct there is an acceleration of the fluid in the cell and when it is connected with the higher pressure duct then there is deceleration of the fluid. Hence, one part of the entering fluid pumps or compresses another part of the same fluid to a higher pressure while that part which has performed the work leaves at reduced pressure.

It is possible to allow the fluid to expand by increasing its speed before leaving the pressure exchanger. In such a case of course, the pressure at which the fluid leaves the exchanger will be lower than it would otherwise have been.

Although in the aforedescribed embodiments not more than two groups of ducts are provided at each end of the cell wheel for the ingoing or for the outgoing fluids, it is nevertheless possible without basically modifying the process or the apparatus, to provide in the end plates, more than two groups of ducts separated by partition walls. Such an apparatus in accordance with the present invention is then capable of taking in fluids at more than two pressure levels and delivering fluids at more than two different pressure levels.

The pressure exchangers described above may be combined in many ways with other similar apparatus or with different apparatus. For example, pressure exchangers may be connected in series and they may be combined with combustion processes or heat introduction or abstraction in different forms. Some of the possible ways of use of the different embodiments of the invention described above will now be set forth. Reference will be made to Figures 6 to 10 which all show applications of the present invention. In Figure 6, 38 is a pressure divider, air entering at 41 and leaving at 42 at a higher pressure and at 43 at a lower pressure. At 40 is a combustion chamber, or other heating means, in which the air entering at 42 is heated, leaving at an elevated temperature through the duct 45. The higher and lower pressure gases are brought together through the ducts 44 and 45 in a pressure equalizer 39, from which the gases are discharged into duct 46, at a pressure higher than prevails at the intake 41. The compressed hot gas at 46 may be utilized to expand through a nozzle and produce thrust, or it may be led to an expansion machine, e.g. a turbine 47, through a duct 48. Useful work is produced and the exhausted gas leaves at 49.

The gas leaving the pressure exchanger 39 may be abducted therefrom, according to an alternative scheme, at two pressure levels; one of which, at 49A, is ambient pressure. This facilitates scavenging and results in a higher efficiency than if the exhaust pressure was above ambient pressure.

It is also possible as a further alternative to take in at 70 ambient air e.g. in order to be able to increase the temperature at 45 without raising the resulting temperature of the pressure exchanger 39.

Supercharging and scavenging of an internal combustion engine for instance two stroke engines, can also be performed by pressure exchangers in a manner based upon the Figure 6 layout. In fact, apparatus taking in gas and delivering it at a higher temperature can be supercharged by a pair of pressure exchangers arranged with the apparatus in place of the combustion chamber 40 of Figure 6. Thus any process such as the firing of a boiler and certain chemical processes can be supercharged as described. The device to be supercharged may advantageously be by-passed by a conduit provided with control means 71 by which it may be throttled or entirely closed.

In arrangements in which pressure exchangers are applied in connection with a turbine, or other expansion machine, to produce mechanical work, the turbine may be so positioned so that unheated gases only flow through it. This arrangement is illustrated in Figure 7 where the turbine 47 is situated in the gas flow between the pressure divider 38 and equalizer 39. The gas enters the turbine at 43 and leaves at 44, with a corresponding drop in pressure. Since the combustion products do not pass in this case through the turbine the arrangement would permit the use of fuels normally excluded because of damage they are expected to cause to the turbine e.g. pulverised coal may be burnt in the combustion chamber 40.

In another alternative arrangement the turbine 47 may work in parallel with the pressure divider, as in Figure 8 the air entering into the turbine in a parallel branch of 41, and leaving at lower pressure. The air leaving the turbine is taken into the pressure equalizer at the same pressure as prevailing in conduit 44, or at a similar pressure level.

The high pressure fluid supplied to a pressure exchanger may be hot gas exhausted from a compressor-turbine set, which it leaves at a pressure above that of the ambient air as is the case in a jet engine. Such gas entering the pressure exchanger, e.g. at 20 on Figure 4, together with low pressure fluid in the form of ambient air entering at 19, can compress the latter and supply gas into a receiver at higher pressure than that of the ambient air. After this the gases can be made to expand through a nozzle orifice (e.g. 26, Figure 4), into the surrounding atmosphere to produce reactive thrust. By such a combination the propulsive efficiency and the thrust are raised, especially at the lower range of speeds at which the engine is moved through the atmosphere.

A schematic arrangement for the purpose described is shown on Figure 9. A compressor 50 is driven by turbine 51 with the combustion system 52 in between. The air enters at 53, leaves the compressor at 54, and after taking up heat enters the turbine at 55, from which it emerges somewhat expanded at 56 and enters the pressure equalizer 57. An ambient air supply is taken in at 58, is compressed in the pressure equalizer and both streams leave together at 59, producing thrust. Alternatively of course, the stream may be expanded in a second turbine as in a turbo-propeller engine. In another arrangement two streams may leave from a pressure exchanger substituted for the equalizer 57.

A further alternative scheme is shown in Figure 10. Here the air entering the compressor 60 at 61, and leaving same at 62, is heated in a combustion system for example at 63 and is led into the pressure equalizer 64 at 65. Air is also taken into the equalizer at 66. The gas leaves the pressure equalizer at 67, enters turbine 68, which drives the compressor and emerges therefrom at 69. If the production of jet thrust is the purpose, then this is produced by the flow at 69; if work on a shaft is to be produced, then this can also be achieved in well known manner. In this alternative the temperature at 65 can be very high, since it is brought down to an

acceptable figure by dilution with the incoming air at 66. Thus neither the turbine 68 nor the pressure equalizer are exposed to excessive temperatures.

This method of reducing the temperature of a gas, without excessive loss of mechanical work available, by diluting a hot gas with a cooler one in a pressure exchanger, is generally applicable. For instance, it makes possible a powerful reheat between the turbine 51 on Figure 9 and the pressure exchanger, or after the pressure exchanger, by which the thrust will be increased further. Since the pressure exchanger receives also cool ambient air, the high reheat temperature will cause no harm in the device.

The fluids employed in pressure exchangers according to the invention may be gaseous or liquids or both combined. It is expected to be possible for instance, to pump by a gas of higher pressure, liquid from a lower pressure to a higher pressure or it can be accelerated to a higher speed.

What I claim is:

1. A pressure exchanger in which gas compression and gas expansion proceed simultaneously comprising a first cylindrical structure having cells around the periphery thereof, a second structure including an end plate adjacent one axial end of said first structure, means for effecting relative rotation between said first and second structures, a plurality of segmental outlet ports in said end plate, together occupying substantially the whole circumferential extent of said end plate and each outlet port being of such circumferential width that in the design conditions of operation wave action initiated within the cells by a leading port edge affects the complete axial length of a cell in both forward and reverse direction, and wall sections in said end plate between adjacent ports each of substantially the same circumferential width as a cell.

2. A pressure exchanger as claimed in claim 1 and comprising a single inlet duct forming part of said second structure and communicating with said cells at their ends opposite said one end.

3. A pressure exchanger as claimed in claim 1, in which said second structure includes an additional end plate adjacent the axial end of said first structure opposite to said one end, at least one segmental inlet port being provided in said additional end plate, the port width occupying substantially the whole circumference of the additional end plate.

4. A pressure exchanger as claimed in claim 3, in which a plurality of said inlet ports are provided, wall sections in said additional end plate between adjacent ports each being of substantially the same circumferential width as a cell and in which said inlet and outlet ports are staggered circumferentially.

5. Apparatus comprising first and second pressure exchangers, said pressure exchangers comprising cells in which gas expands so compressing another part of that gas with which it is in direct contact, said cells forming a continuous boundary for the gas flowing through them and extending in the direction of gas flow ducting to lead gas to and from the cells at different pressure levels, and means to effect relative motion between the cells and the ducting, the first pressure exchanger receiving gas through said ducting at a first pressure level and discharging said gas through said ducting at two pressure levels one higher and the other lower than said first level, said apparatus further comprising a gas heating system receiving gas at said higher pressure level directly from the first pressure exchanger and discharging hot gas, and the second pressure exchanger receiving said hot gas directly from the heating system together with gas discharged at said lower pressure level from the first pressure exchanger and discharging a stream of hot gas.

6. A heat engine incorporating pressure exchanger apparatus as claimed in claim 5 and in which an expansion machine is provided through which said stream is expanded to perform useful work.

7. A heat engine incorporating pressure exchanger apparatus as claimed in claim 5 in which said second pressure exchanger has three inlet ducts, a first duct to receive said hot gas from said gas heating system, a second duct to receive gas discharged at said lower pressure level from said first pressure exchanger and a third duct to receive a further gas supply at a low temperature compared with said hot gas whereby greater heat may be introduced via the heating system than would otherwise be permissible.

8. A heat engine incorporating pressure exchanger apparatus as claimed in claim 5 modified in that an internal combustion engine required to be supercharged at said higher pressure level and scavenged is substituted for said heating system.

9. A heat engine incorporating pressure exchanger apparatus as claimed in claim 5 in which said second pressure exchanger has two outlet ducts, a first duct for said hot gas stream and a second duct for a second output stream of gas at a lower pressure level than said discharged hot gas.

10. A pressure exchanger in which gas compression and gas expansion proceed simultaneously, comprising a first cylindrical structure having cells around the periphery thereof, an inlet duct through which fluid enters the cells at one end of said first cylindrical structure, a second structure including an end plate in juxtaposed relation to the other end of said first cylindrical structure, means for effecting relative rotation between said first and said second structures, the fluid in the cells of said first structure being subjected to acceleration and deceleration impulses to create pressure differences between opposite ends of the cells in the said first cylindrical structure, said end plate having a plurality of outlet ports and ducting communicating with said outlet ports for the passage of fluid from the cells of said first cylindrical structure to alternate outlet ports and ducts at pressure levels respectively above and below the pressure of the fluid in the inlet duct of said first cylindrical structure.

11. A pressure divider in which gas compression and gas expansion proceed simultaneously, comprising a cylindrical structure having cells around the periphery thereof, an inlet duct through which fluid enters the cells at one end of said cylindrical structure, an end plate in juxtaposed relation to the other end of said cylindrical structure, means for effecting relative rotation between said end plate and said cylindrical structure, said end plate having therein a plurality of outlet ports communicating with said cells, wall sections in said end plate separating said outlet ports, the width dimensions of said outlet ports, wall sections and cells being of such magnitudes relative to each other that in the designed conditions of operation the fluid from said inlet duct is delivered to alternate outlet ports via said cells at pressure levels respectively above and below that of the fluid in said inlet duct.

12. A pressure divider in which gas compression and gas expansion proceed simultaneously, comprising a first cylindrical structure having cells around the periphery thereof, an inlet duct through which fluid enters the cells at one end of said first cylindrical structure, a second structure including an end plate in juxtaposed relation to the other end of said first cylindrical structure, means for effecting relative rotation between said first and second structures, said end plate having therein a plurality of segmental outlet ports which together occupy substantially the whole circumferential extent of said end plate, each outlet port being of such circumferential width that in the design conditions of operation the fluid in the cells is subjected to acceleration and deceleration impulses initiated by the leading and trailing edges of the outlet ports which impulses create pressure differences between opposite ends of said cells, ducting communicating with

said outlet ports, wall sections separating said outlet ports in said end plate, said wall sections being of substantially the same circumferential width as said cells, the width dimensions of said outlet ports, wall sections and cells being of such magnitudes relative to each other that in the designed conditions of operation the fluid from said inlet duct is delivered to alternate outlet ports via said cells at pressure levels respectively above and below that of the fluid in said inlet duct.

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